

**CHEMICAL ENGINEERING
LABORATORY EQUIPMENT**

Design · Construction · Operation

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CHEMICAL ENGINEERING LABORATORY EQUIPMENT

Design · Construction · Operation

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PREFACE

This book is an outgrowth of *Unit Operations Laboratory Equipment* which was published in mimeographed form in two editions, in 1938 and 1940. The principal object in the preparation of these earlier editions was to pool the experiences of men in various chemical engineering departments in the design and construction of equipment particularly suited for instructional purposes. It was not our intention to provide a laboratory manual of the usual type because we did not believe that a standardized manual was desirable, for when the student follows minute and explicit directions he usually fails to learn the principals that the experiment is designed to demonstrate. Although brief experimental procedures and typical student results were included with each report, this was done only to indicate the performance of the equipment.

Although the earlier editions were prepared principally for those who were interested in chemical engineering education, it was soon found that the information presented was of equal value to those engaged in industrial research and development, and even in plant operation. As a result of many valuable suggestions, we decided to not only include considerable new material in the present book, but also to modify the material of the earlier editions. Although the general form of the earlier editions has been retained, added emphasis has been placed on those portions of each report dealing with the presentation of the data and the calculations involved. It is believed that this will be of value to students, to teachers, and particularly to men in industry.

A large part of the material for this book is based on reports submitted by various men engaged in the teaching of chemical engineering. Most of these reports were completely rewritten to conform to a standard method of presentation, new drawings were prepared, and much of the material pertaining to the presentation of results and calculations was expanded in order to clarify the method of handling the data.

We are still of the opinion that students should not be furnished with detailed instructions for the operation of equipment or for the presentation of experimental results; and most of those who have contributed reports are of the same opinion. Furthermore, even if detailed instructions were desirable, it would be impossible to prepare a satisfactory chemical engineering laboratory manual of the type commonly used in chemistry courses because of the wide differences between similar types of equipment in different laboratories, and the differences of opinion

among instructors of chemical engineering laboratory courses as to the most desirable experiments to be performed. Therefore, the typical experiments that are presented in this book are offered merely as examples of methods for carrying out experiments and handling the data that are obtained. It is believed that students, instructors, and men in industry will be aided in the performance of their own experimental work by the study of the techniques and methods used by others.

It often becomes necessary in instructional and industrial chemical engineering laboratories to construct a piece of equipment for a definite type of experimental work or to modify an existing piece of equipment to make it more suitable for the desired purpose. In this book, we have presented a number of designs of equipment for various purposes and have given detailed methods of construction. Although the details of construction are sufficiently complete so that each piece of equipment can be built to the dimensions given, the designs may easily be modified to suit individual needs or preferences.

We wish to express our appreciation to those who have contributed reports for this book as well as to the many others who have given us valuable suggestions. We particularly wish to acknowledge the valuable assistance of Mr. Howard Wilson and Miss Juliette Laflamme: Mr. Wilson prepared many of the drawings, and Miss Laflamme did an excellent job of typing a difficult manuscript.

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SECTION 1

FLOW OF FLUIDS

INTRODUCTION

Fluid-flow experiments usually involve the determination of pressure drops in pipes and fittings and the correlation of the experimental results with values obtained from the literature or calculated by means of accepted equations. As pressure-drop experiments involve the measurement of flow rates, the calibration of flow-rate instruments, such as orifice and Venturi meters and pitot tubes, is usually included as part of the experiment. It is customary to express the results of such a calibration in the form of an equation and to compare the equation so obtained with the generally accepted one for that type of instrument.

Obviously, fluid-flow experiments are not limited to flow through pipe and fittings for the same technique is involved in measuring the pressure drop through catalytic chambers, filter beds, heat exchangers, absorption towers, or any other system through which a fluid is flowing. The correlation of the results, however, is usually more difficult because such systems are more complex.

THE MEASUREMENT OF PRESSURE DROP

The pressure drop across any two points in a system may be measured by means of various kinds of liquid manometers or other types of gages, depending upon the magnitude of the pressure drop, the total pressure involved, and the desired accuracy of the results. If the total pressure is not very great, the pressure at any point in a liquid system can be measured most easily and accurately by means of an open-end glass piezometer tube; if the pressure is too great to permit the use of this method, a U-tube manometer connected between the two points across which the pressure drop is being measured is ordinarily used. Although U-tube mercury manometers are most common, low pressure drops are more accurately measured by means of inclined mercury manometers, inverted U-tube manometers of the air-liquid type, or U-tube manometers in which the mercury is replaced by an immiscible liquid of lower specific gravity, such as carbon tetrachloride. The use of pressure gages of the Bourdon type is ordinarily limited to those systems where the pressure is high and the pressure drop is large.

Pressure-Tap Openings. The accuracy of pressure-drop measure-

ments depends largely on the care taken in constructing and locating the pressure tap openings. The following precautions should be observed:

1. The size of the opening should be small compared with the cross sectional area of the duct. Theoretically, the smaller the diameter of this opening the more accurate are the results.⁸ However, the actual size of the opening is determined largely by practical considerations. If the opening is too small, it may become plugged; this is particularly true in systems using water and steel pipe, for rusting and scale formation are quite rapid. Ordinarily, a $\frac{1}{8}$ -inch hole is satisfactory for most purposes.

2. The opening should be perpendicular to the wall of the pipe and flush with the inside surface, and the inner edge should be free from burrs or other irregularities.* Furthermore, the inner surface of the pipe where the hole is drilled should be substantially smooth for a distance equal to at least two pipe diameters on each side of the opening.

3. Usually only one opening is drilled at a given location, but sometimes several openings are drilled in the same cross section and connected to a common manifold to form a piezometer ring. This is done with the idea of automatically averaging differences in pressure that may exist at the several openings. However, if appreciable differences in pressures do exist, it is evident that the flow is not parallel to the walls of the pipe and, therefore, the pressure taps are either improperly located or improperly constructed. In such cases, the average reading is of little value, for it does not indicate the true condition of flow. On the whole, a piezometer ring is of questionable value and the use of a properly constructed single opening is preferable. However, a piezometer ring is sometimes used because it reduces the possibility of complete plugging, but this is poor justification for its use.

4. The pressure-tap openings should be located sufficiently far from any valve, elbow, or other fitting so as to be beyond the region of eddies, whirls, or any other irregularities of flow caused by the valve or fitting. This distance will vary with the magnitude of the disturbance, but for practical purposes a distance of 30 diameters downstream and 5 diameters upstream is sufficient for even conditions of maximum disturbance. Where it is not feasible to locate the downstream tap at a sufficient distance, straightening vanes may be employed; a simple and convenient flow-straightening device is a bundle of small-diameter tubes fastened rigidly in the conduit and parallel to its axis.

* The construction of piezometer (pressure-tap) openings is described on pages 27 and 53.

FLUID-FLOW EQUATIONS

The application of the law of conservation of energy to the flow of fluids under steady conditions results in the important equation, generally known as *Bernoulli's theorem*, which may be written as^{1, 12}

$$x_1 + p_1 v_1 + \frac{u_1^2}{2g_c} + \int_1^2 p dv + W = x_2 + p_2 v_2 + \frac{u_2^2}{2g_c} + F \quad (1.1)$$

or as

$$W - \int_1^2 v dp = x_2 - x_1 + \frac{u_2^2 - u_1^2}{2g_c} + F \quad (1.2)$$

since

$$\int_1^2 p dv + p_1 v_1 - p_2 v_2 = - \int_1^2 v dp \quad (1.3)$$

In these equations:

x = height above an arbitrary horizontal datum plane, ft.

u = velocity, ft./sec.

g_c = conversion factor* = 32.174 (lb. mass \times ft.)/(lb. force \times sec.²).

p = absolute pressure, lb./ft.²

v = specific volume of fluid, ft.³/lb. = $\frac{1}{\text{density}} = \frac{1}{\rho}$.

W = work input from surroundings, ft.-lb./lb.

F = friction loss, ft.-lb./lb., or ft. of head.

The friction loss F is a function of the flow rate and the length and size of the duct, and depends upon the number and kind of valves and fittings in the line.

Flow Through Straight Pipe. For straight circular pipes, the relationship between F and the conditions of flow is expressed by the *Fanning equation*⁴

$$F = \frac{4fLu^2}{2g_c D} = \frac{4fLG^3}{2g_c D\rho^3} \quad (1.4)\dagger$$

or

$$\Delta p = F\rho = \frac{4fL\rho u^2}{2g_c D} = \frac{4fLG^2}{2g_c D\rho} \quad (1.5)$$

* g_c is the factor used to convert absolute units to English engineering units; it has the same numerical value as the standard acceleration due to gravity.

† In many books, particularly in civil and mechanical engineering, this equation is given as $F = \frac{fLu^3}{2g_c D}$, which results in values of f four times as great as in (1.4) and subsequent equations.

where

Δp = pressure drop, lb./ft.²

f = the Fanning friction factor, dimensionless.

L = length of pipe, ft.

D = diameter of pipe, ft.

G = mass velocity, lb./ (sec. \times ft.²).

ρ = density, lb./ft.³

The Fanning friction factor, in turn, is a function of the dimensionless parameter $\frac{Du\rho}{\mu}$, known as the *Reynolds' number*, Re . This function is not continuous and a plot of f vs. Re on log-log paper (Figure 1) consists of two distinctly different curves. When the Reynolds' number is less than approximately 2100, flow is of the viscous, or laminar, type and the friction factor curve is a straight line with a slope of minus one; the equation of this curve is

$$f = \frac{16}{Re} \quad (1.6)$$

This relationship, which is independent of the roughness of the pipe, is derived from the *Hagen-Poiseuille equation* for streamline flow in circular pipes²

$$\Delta p = \frac{32Lu\mu}{g_c D^2} \quad (1.7)$$

where

Δp = pressure drop due to friction, lb./ft.²

μ = viscosity of the fluid, lb./ (ft. \times sec.).

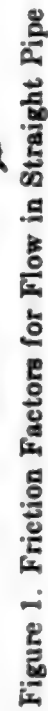
Equating (1.5) and (1.7):

$$\Delta p = \frac{4fL\rho u^2}{2g_c D} = \frac{32Lu\mu}{g_c D^2}$$

or

$$f = \frac{16\mu}{Du\rho} = \frac{16}{Re}$$

When the Reynolds' number is above the critical value of 2100, the flow is usually turbulent, although there is a critical region between Reynolds' numbers of 2100 and about 3500 in which flow may be either viscous or turbulent, or partially viscous and partially turbulent.^{9, 10, 18}



The relationship between the friction factor and Reynolds' number is much more complex for turbulent flow than it is for viscous flow, and in the turbulent region the frictional loss depends greatly upon the roughness of the pipe. Many equations have been proposed for the relationship between f and Re in the turbulent region;* most of these equations are of the form^{6, 13, 14, 15}

$$f = a + bRe^n \quad (1.8)$$

where

a , b , and n are constants.

Perhaps the most widely used of these equations are those developed by Koo;⁶ these equations are:

1. *For smooth clean tubes, such as glass, lead, and copper*

$$f = 0.00140 + 0.125 Re^{-0.32} \quad (1.9)$$

2. *For clean iron and steel pipe*

$$f = 0.00307 + 0.1886 Re^{-0.36} \quad (1.10)$$

The curves of Figure 1 for the turbulent region are based on these equations.

Flow Through Valves and Fittings. When a valve or fitting is inserted in a pipe, or the pipe is enlarged or contracted, the turbulence of the fluid stream is increased and this results in a permanent loss of head. However, because of the wide variety of fittings and the variations in the individual fittings of the same type, no satisfactory method has been developed for calculating this loss of head.

Enlargement and contraction losses are usually calculated by means of the equations:

1. *For enlargement*

$$F_e = \frac{(u_2 - u_1)^2}{2g_c} \quad (1.11)$$

2. *For contraction*

$$F_c = K_c \frac{u_2^2}{2g_c} \quad (1.12)$$

where

u_2 and u_1 are the velocities in feet per second in the smaller and

* An excellent comparison of the various equations has been made in the recently published book, *Flow of Water in Pipe and Pipe Fittings*, by J. R. Freeman, American Society of Mechanical Engineers, New York, 1941, pp. 94-102.

larger cross sections, respectively, and K_c is a coefficient whose value depends upon the ratio of the areas of the two sections (Figure 2).¹¹

For valves, elbows, tees, return bends, etc., the resistance of the fitting is most commonly expressed in terms of the equivalent feet of straight pipe which would have the same resistance as the fitting. Figure 3 was prepared in this manner. In a second method, the loss in fittings is expressed in the same way as contraction losses, by the equation

$$F_f = K_f \frac{u^2}{2g_c} \quad (1.13)$$

where K_f is an empirically determined coefficient which varies with the size of the pipe and the Reynolds' number.⁷

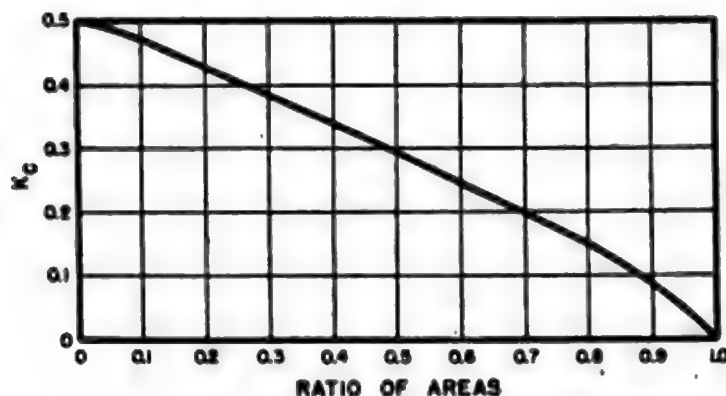


Figure 2. Contraction Coefficient K_c in the Formula $F_c = K_c \frac{u_1^2}{2g_c}$.

CALCULATION OF FRICTION FACTOR FROM EXPERIMENTAL RESULTS

The calculation of the friction factor from experimental data involves the use of the Bernoulli and Fanning equations (Equations 1.1 or 1.2 and 1.4 or 1.5). The treatment for the flow of liquids is somewhat simpler than for the flow of gases except when the pressure drop of the gas is small.

Calculations for Liquids. When a liquid is flowing through a pipe of uniform cross-sectional area, Bernoulli's equation can be simplified because of the following conditions: (1) $W = 0$, if there is no pump in the section under consideration, (2) $x_1 = x_2$, either because the pipe is horizontal, the gages are at the same level, or the manometer connected between the two points does not measure the difference in hydrostatic heads, (3) $v_1 = v_2$, $\int p dv = 0$, and $\int v dp = v(p_2 - p_1)$,

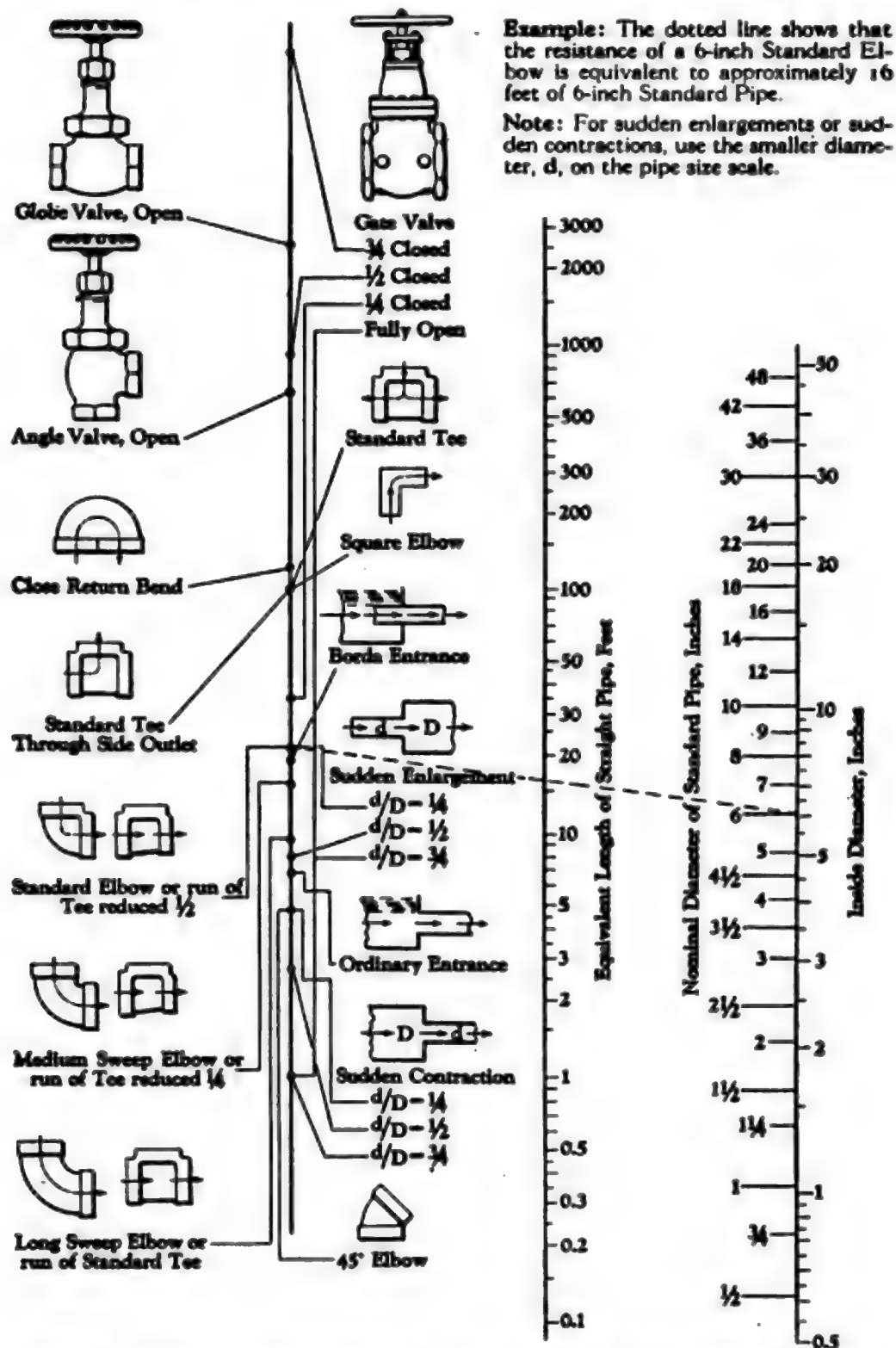


Figure 3. Resistance of Valves and Fittings to Flow of Fluids (Courtesy of Crane Co.)

because a liquid is practically incompressible, and (4) $u_1 = u_2$. Under these conditions, Equation (1.2) becomes

$$F = v\Delta p = \frac{\Delta p}{\rho} \quad (1.14)$$

where

$$\Delta p = \text{pressure drop, lb./ft.}^2 = (p_1 - p_2).$$

and from Equations (1.4), (1.5), and (1.14),

$$f = \frac{Fg_c D}{2Lu^2} = \frac{v\Delta pg_c D}{2Lu^2} = \frac{\Delta pg_c D}{2L\rho u^2} \quad (1.15)$$

Calculations for Gases. For the flow of gases, $W = 0$ and $x_1 = x_2$ under the usual conditions of operation, but as a gas is a compressible fluid its velocity and specific volume change along the pipe as the pressure changes.

For Small Pressure Drops. If the pressure drop is not very great $-\int_1^2 v dp$ is approximately equal to $(p_1 - p_2) v_{av}$ or $v_{av}\Delta p$, and $\frac{u_2^2 - u_1^2}{2g_c}$ is sufficiently small so that it may be neglected. The value of f , therefore, can be calculated from the equation

$$f = \frac{Fg_c D}{2Lu_{av}^2} = \frac{v_{av}\Delta pg_c D}{2Lu_{av}^2} = \frac{\Delta pg_c D}{2L\rho_{av} u_{av}^2} \quad (1.16)$$

where v_{av} , ρ_{av} and u_{av} are the average values of the specific volume, the density, and the velocity, respectively.

This equation is usually recommended where $\Delta p < 0.10 p_1$, although for isothermal flow of gases it is probably within the accuracy of the friction data¹⁶ when $\Delta p \leq 0.50 p_1$.

A slightly different equation can be derived if the flow is substantially isothermal and if ideal gas laws are assumed, for

$$p_1 v_1 = p_2 v_2$$

and

$$\int_1^2 p dv = -\int_1^2 v dp = \int_2^1 v dp = RT \int_2^1 \frac{dp}{p} = 2.303RT \log_{10} p_1/p_2$$

Therefore,

$$F = -\int_1^2 v dp = 2.303RT \log_{10} p_1/p_2 = \frac{4fLu_{av}^2}{2g_c D}$$

and

$$f = \frac{2.303RTg_c D}{2Lu_a^3} \log_{10} \frac{p_1}{p_2} \quad (1.17)$$

For Large Pressure Drops. When the pressure drop is too great to permit the use of the simpler equations, the work of expansion and the increase in kinetic energy must be considered.

The general equation for the flow of a gas in a conduit may be written as

$$v dp + \frac{u du}{g_c} + \frac{2fu^2 dL}{g_c D} = 0 \quad (1.18)$$

Dividing through by u^2 and multiplying by $2g_c$

$$\frac{2g_c v dp}{u^2} + \frac{2 du}{u} + \frac{4f dL}{D} = 0 \quad (1.19)$$

If

w = weight of fluid per unit time.

A = cross-sectional area of the duct.

$v = RT/p$.

then

$$u = \frac{wv}{A} = \frac{wRT}{Ap}$$

Therefore,

$$\frac{2g_c v dp}{u^2} = \frac{2g_c (RT/p) dp}{\left(\frac{wRT}{Ap}\right)^2} = \frac{2g_c A^2 p dp}{w^2 RT} \quad (1.20)$$

and Equation (1.19) becomes

$$\frac{2g_c A^2 p dp}{w^2 RT} + \frac{2 du}{u} + \frac{4f dL}{D} = 0 \quad (1.21)$$

Integrating between limits

$$\frac{2g_c A^2}{w^2 RT} \int_{p_1}^{p_2} p dp + 2 \int_{u_1}^{u_2} \frac{du}{u} + \frac{4f}{D} \int_0^L dL = 0$$

and

$$\frac{g_c A^2}{w^2 RT} (p_2^2 - p_1^2) + 2 \log_{10} \frac{u_2}{u_1} + \frac{4fL}{D} = 0$$

or

$$p_1^2 - p_2^2 = \frac{w^2 RT}{g_c A^2} \left[2 \log_e \frac{u_2}{u_1} + \frac{4fL}{D} \right] \quad (1.22)$$

and, where the velocity term is negligible compared to the other terms

$$p_1^2 - p_2^2 = \frac{4fLw^2 RT}{g_c A^2 D} \quad (1.23)$$

and

$$f = \frac{g_c A^2 D}{4Lw^2 RT} (p_1^2 - p_2^2) \quad (1.24)$$

In these equations

- f = the Fanning friction factor, dimensionless.
- g_c = conversion factor = 32.174 (lb. mass \times ft.)/(lb. force \times sec.²).
- A = cross-sectional area of duct, ft.²
- D = diameter of duct, ft.
- L = length of duct, ft.
- w = flow rate, lb./sec.
- R = gas constant, ft.-lb./(lb. \times °R.).
- T = absolute temperature, °R.
- p_1 = initial pressure, lb./ft.²
- p_2 = final pressure lb./ft.²

Accuracy of the Equations. When experimental results for friction losses are compared with values calculated from theoretical or empirical equations or with values reported in the literature, the discrepancy is frequently much larger than that due to experimental error. This is particularly true of expansion and contraction losses and losses in valves and fittings, for the agreement between the observed and calculated results for the frictional loss in straight pipe is usually satisfactory.

The equation of Koo for smooth tubes (Equation 1.9) expresses the results of a large number of experiments with an accuracy of ± 5 per cent, and the equation for commercial steel pipe (Equation 1.10) is accurate to approximately ± 10 per cent. Of course, if the pipe is excessively rough, the observed values may be much higher than those calculated. Experiments with both naturally and artificially roughened pipe have led to numerous attempts to develop roughness factors¹⁷ to be applied as corrections to values calculated from the equations for smooth tubes or commercial steel pipe, but such factors are of little importance because there is no satisfactory way in which to measure roughness.

The calculated values for expansion and contraction are frequently in error, for the magnitude of such losses depends upon the abruptness of the change in cross-sectional area and the type of fittings used to make the reduction. In general, the more gradual the reduction, the lower the loss. In any case, however, it seems desirable to modify Equation (1.11) to read

$$F_s = K_s \frac{(u_2 - u_1)^2}{2g_s} \quad (1.25)$$

where K_s depends upon the type of fitting used to make the enlargement.⁷

When calculating the expansion or contraction loss from experimental data, it must be remembered that a manometer across the junction of the two pipes measures not only the permanent loss in the fitting but also the differences in the velocity heads, and that in the case of contraction the difference in the velocity heads must be added to the reading while in the case of expansion it must be subtracted.

The experimental values for losses in valves and fittings often differ by several hundred per cent from the generally accepted values. This is to be expected, for: (1) no two valves or fittings are exactly alike, and small differences in construction may make a large difference in the resistance, and (2) the feet of pipe equivalent to a fitting—or the value of K_f in Equation (1.13)—is not constant but varies with the size of the fitting and with the velocity. Although variation with velocity is usually not excessive at velocities greater than about 5 feet per second for water, the equivalent feet per fitting usually drops rapidly as the velocity is decreased below this value, although in some cases it becomes greater.

THE MEASUREMENT OF FLUID-FLOW RATES

The most common laboratory flow meters are the pitot tube, the orifice meter, and the Venturi meter. The pitot tube is more extensively used for gases than for liquids, but Venturi and orifice meters are employed for both types of fluids. The Venturi meter is much more expensive than the orifice meter but, because the permanent head loss is much less in a Venturi than in an orifice, the Venturi is preferable for permanent installations where power costs must be considered. For laboratory purposes, however, where equipment is used only intermittently and for periods of short duration, the Venturi meter is seldom justified except for instructional purposes or where the high pressure

drop through an orifice must be avoided. A very satisfactory, and yet inexpensive, Venturi meter can be constructed as shown in Figure 4;

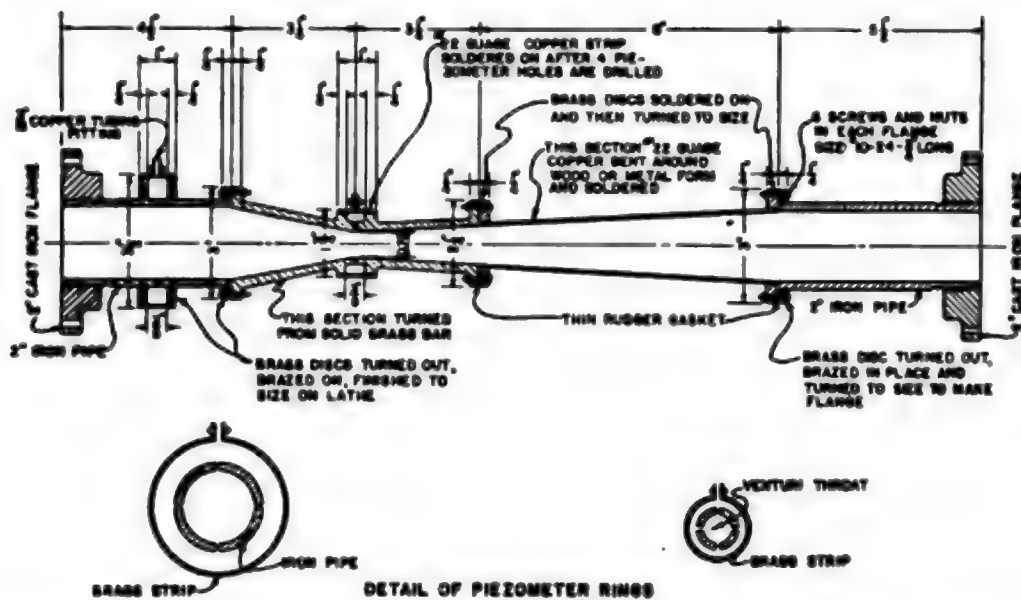


Figure 4. Details of Construction of a 2-inch Venturi Meter with a 0.75-inch Throat—University of New Hampshire

and an inexpensive orifice meter is easily made by inserting a thin orifice plate between flanges or between the two halves of a gasket-type union, or it can be made as shown in Figure 5.

A general equation that is applicable for all three types of meters—pitot tube, orifice meter, and Venturi meter—can be written in the form

$$u = K\sqrt{2g_c\Delta H} \quad (1.26)$$

where

- u = velocity.
- K = a constant.
- g_c = conversion factor.
- ΔH = manometer reading.

The calibration data are most conveniently plotted on log-log paper. for the resulting curves are straight lines with slopes of 0.5.

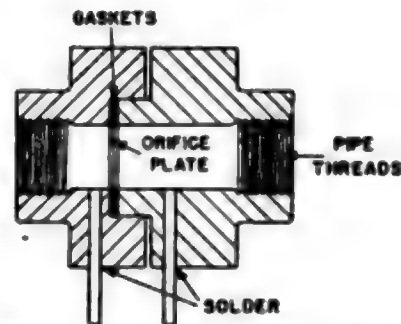


Figure 5. Design of a Small Orifice Meter—University of North Dakota

Pitot Tube. The value of K in Equation (1.26) for the pitot tube is practically 1 when ΔH is expressed in feet of fluid; therefore, for this type of flow meter, the equation becomes

$$u = \sqrt{2g_c\Delta H} \quad (1.27)$$

where

u = velocity in ft./sec.

g_c = conversion factor = 32.174 (lb. mass \times ft.)/(lb. force \times sec.²).

ΔH = ft. of fluid.

Orifice Meter. The equation for the orifice meter is usually written as

$$\sqrt{u_2^2 - u_1^2} = C_o\sqrt{2g_c\Delta H} \quad (1.28)$$

where

u_2 and u_1 = velocity, in ft./sec., through the orifice and the pipe, respectively.

ΔH = pressure drop in ft. of fluid.

g_c = conversion factor = 32.174 (lb. mass \times ft.)/(lb. force \times sec.²).

C_o = a constant, which usually has a value of about 0.61; however, this value varies somewhat depending upon the ratio of the orifice to pipe diameters, the size of the pipe, and the Reynolds' number.³

Venturi Meter. The equation for the Venturi meter is generally written as

$$\sqrt{u_2^2 - u_1^2} = C_v\sqrt{2g_c\Delta H} \quad (1.29)$$

where

u_2 and u_1 = velocity in ft. per sec. through the Venturi throat and the pipe, respectively.

C_v = a constant which, like the discharge coefficient for the orifice meter, varies slightly with the ratio of the Venturi throat to pipe diameters, the size of the pipe, and the Reynolds' number. For most practical purposes, C_v can be taken as 0.98.

and

ΔH and g_c have the same units as in Equation (1.28).

In plotting calibration data, it is more convenient to transform Equation (1.28) or (1.29) into the form of Equation (1.26). This can be done by eliminating u_2 .

Since

$$\frac{u_2}{u_1} = \frac{D_1^2}{D_2^2} \quad \text{or} \quad u_2 = u_1 \left(\frac{D_1}{D_2} \right)^2$$

the general equation for orifice and Venturi meters— $\sqrt{u_2^2 - u_1^2} = C\sqrt{2g_c\Delta H}$ —can be written as

$$\sqrt{u_1^2 \left(\frac{D_1}{D_2} \right)^4 - u_1^2} = C\sqrt{2g_c\Delta H}$$

or

$$\sqrt{u_1^2 \left[\left(\frac{D_1}{D_2} \right)^4 - 1 \right]} = C\sqrt{2g_c\Delta H}$$

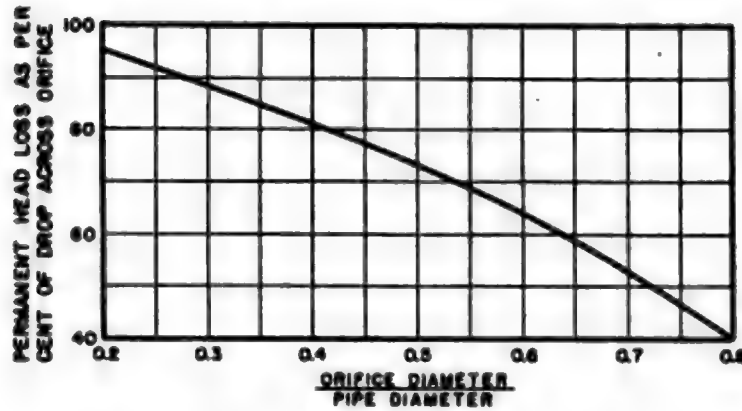


Figure 6. Permanent Head Loss in Orifice

and

$$\begin{aligned} u_1 &= C \frac{1}{\sqrt{\left(\frac{D_1}{D_2} \right)^4 - 1}} \sqrt{2g_c\Delta H} \\ &= K\sqrt{2g_c\Delta H} \end{aligned} \quad (1.30)$$

where

$$K = (C_v \text{ or } C_o) \times \frac{1}{\sqrt{\left(\frac{D_1}{D_2} \right)^4 - 1}} \quad (1.31)$$

Permanent Head Loss in Orifice and Venturi Meters. In using an orifice or a Venturi meter, it must be realized that the measured drop in pressure is due principally to a difference in the velocity heads, and that as the flow returns to normal after passing through the meter most of this drop in head is recovered. However, because of friction,

some of the head is permanently lost. For the orifice meter, the magnitude of this loss, which varies with the ratio of orifice to pipe diameters, is given in Figure 6. The permanent head loss in the Venturi meter is much less—from 10 to 15 per cent of the pressure-drop reading.

NOMENCLATURE

A	= cross-sectional area, ft. ²
C	= discharge coefficient, dimensionless.
D	= diameter of duct, ft.
f	= Fanning friction factor, dimensionless.
F	= friction loss, ft.-lb./lb., or ft. of head.
g_c	= conversion factor = 32.174 (lb. mass \times ft.)/(lb. force \times sec. ²).
G	= mass velocity, lb./(sec. \times ft. ²).
ΔH	= loss in head, ft. of fluid.
K	= constant or coefficient, dimensionless.
L	= length of duct, ft.
p	= absolute pressure, lb./ft. ²
Δp	= pressure drop, lb./ft. ²
R	= gas constant, ft.-lb./(lb. \times °R.).
Re	= Reynolds' number = $Du\rho/\mu$, dimensionless.
T	= absolute temperature, °R.
u	= linear velocity, ft./sec.
v	= specific volume, ft. ³ /lb.
w	= flow rate, lb./sec.
W	= work input from surroundings, ft.-lb./lb.
x	= height above datum plane, ft.
μ	= viscosity, lb./(ft. \times sec.).
ρ	= density, lb./ft. ³

Subscripts

1	= initial condition.
2	= final condition.
av	= average.
c	= contraction, except in the conversion factor g_c .
e	= enlargement.
f	= fitting.
o	= orifice.
v	= Venturi.

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AN APPARATUS FOR VISUAL OBSERVATION OF FLUID FLOW

Designed and Constructed by

The Department of Chemical and Gas Engineering
Johns Hopkins University*

Description. This apparatus, which is designed to illustrate the original Reynolds' experiment, consists essentially of a large calming tank and a Pyrex flow tube. The unit is arranged so that flow tubes of different diameters can be used. The determination of the critical Reynolds' number is made by visual observation of the turbulence of a small thread of dye in the fluid stream and also by pressure drop measurements.^{1, 2, 3}

Figure 1 is a drawing of the assembled apparatus and Figure 2 is an enlarged view of the flow-tube entrance over the distance A-A of Figure 1.

Materials and Cost. The materials used in constructing this apparatus are listed in Table 1; exclusive of the calming tank, their cost amounted to approximately \$43.

Notes on Construction. 1. In order to reduce turbulence in calming tank A to a minimum, $\frac{1}{8}$ -inch holes are drilled in water-distributor line B on the side of the pipe facing the nearest wall of the tank.

2. Wood support C, below the flow tube, is painted white in order to make it easier to observe the characteristics of the colored thread in the fluid stream.

3. The break between viscous and turbulent flow occurs at about the mid-point of flow tube D; therefore, it would be better to make the tubes with a 20-inch straight section between the inlet and the first manometer tap.

4. The distance between manometer taps is 30 inches. This is about the minimum distance for satisfactory manometer readings because the pressure drop at low rates of flow is less than one centimeter of water.

Operation. Water, under a constant head in the calming tank, is passed into the inlet of the flow tube where it meets a fine thread of potassium permanganate solution which is introduced at the center of the stream. The appearance of the colored stream is noted, the flow rate is determined by weighing the quantity of water during a measured interval of time, and the pressure drop across the length of

* Reported by C. E. Thompson and C. F. Bonilla.

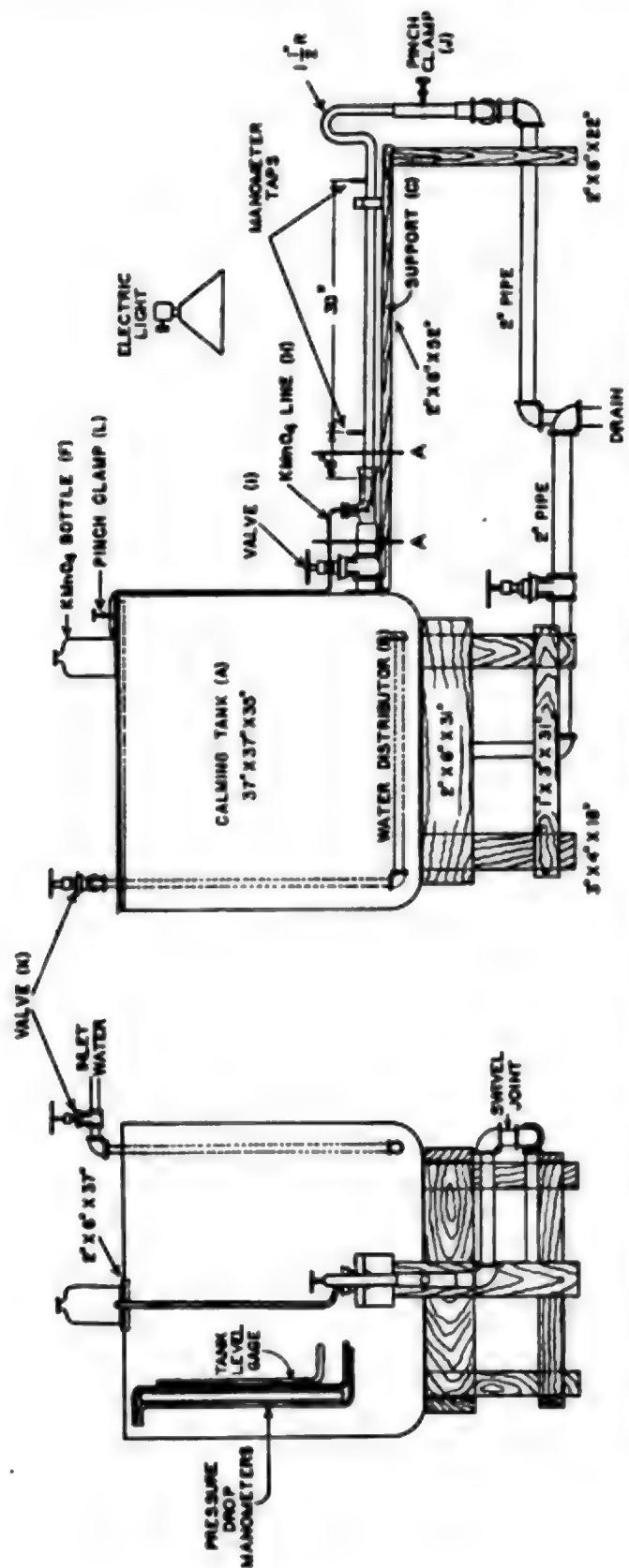


Figure 1. Assembly of Apparatus for Visual Observation of Fluid Flow—Johns Hopkins University

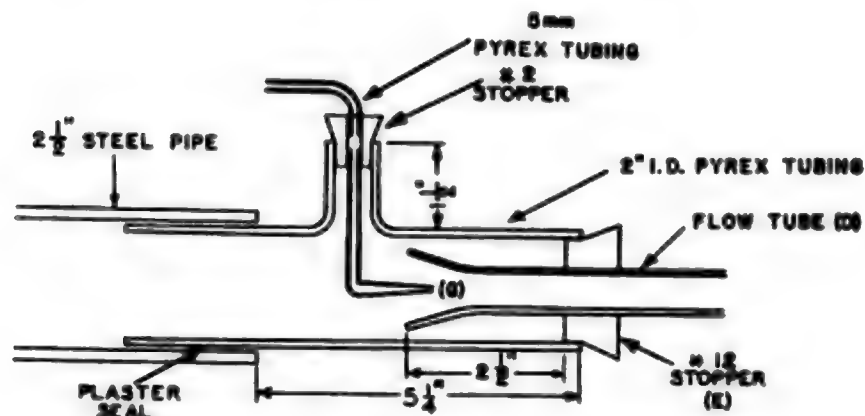


Figure 2. Enlarged View of Flow-tube Entrance

TABLE 1
Bill of Materials for Visual Fluid-Flow Apparatus
 Johns Hopkins University

Quantity	Description	Size	Material
6 in.	Std. pipe	2½ in.	Black wrought steel
8 ft.	" "	2 in.	" " "
5 ft.	" "	½ in.	" " "
1	Std. nipple	2½ in., short	" " "
2	" "	2 in., short	" " "
1	" "	½ in., short	" " "
7	Std. 90° elbow	2 in.	Black malleable steel
2	" " "	½ in.	" " "
1	Std. cap	½ in.	" " "
1	Std. gate valve	2½ in.	Brass
1	" " "	2 in.	"
1	" " "	½ in.	"
8 in.	Tubing	2 in. I.D.	Pyrex glass
5 ft.	"	12 mm.	" "
5 ft.	"	10 mm.	" "
5 ft.	"	8 mm.	" "
10 ft.	"	5 mm.	" "
1	Calming tank	37 x 37 x 35 in.	Steel
3	Stoppers	No. 12	Rubber
1	"	No. 2	"
7 ft.	Tubing	½ in.	"
1	Aspirator bottle	1 liter	Glass
6 ft.	Lumber	3 x 4 in.	Pine
16 ft.	"	2 x 6 in.	"
10 ft.	"	1 x 3 in.	"

the flow tube is measured by means of open-end manometers connected to the taps with rubber tubing.

This operation is carried out at different flow rates and with different-sized tubes. With the fluid head available, the flow in the 8-mm. tube is mainly viscous, the flow in the 12 mm. tube is mainly turbulent, and the flow in the 10 mm. tube may be adjusted for both the viscous and turbulent ranges.

TYPICAL EXPERIMENT

Objects. 1. To study the differences in the flow pattern and the friction head between turbulent and viscous flow.

2. To determine the critical value of the Reynolds' number by visual observation and by friction head measurements.

Procedure. 1. Connect the 8-mm. Pyrex flow tube *D* to the calming tank and securely wire stopper *E* so that it will not be blown out under pressure.

2. Adjust permanganate-discharge orifice *G* to the center of the inlet mouth of the flow tube, and fill aspirator bottle *F* and line *H* with dilute potassium permanganate solution, completely displacing all air from the line.

3. Run water into calming tank *A* until it is at least two-thirds full, partially open 2-inch valve *I*, and then gradually restrict the flow by pinch clamp *J* on the discharge end of the flow tube. Finally, open valve *I* completely in order to avoid eddy currents, and control the flow rate by means of clamp *J*. During the experiments, adjust water-inlet valve *K* so that the water level in the calming tank is maintained constant. Measure and record the temperature of the water several times during the experiments.

4. Carefully fill the manometer and leads with water, making certain that all air is removed from the lines.

5. Adjust the flow through the tube so that the rate is well below the critical velocity and then start the flow of the permanganate solution, controlling its rate by means of pinch clamp *L* so that there is a very fine colored thread in the center of the main stream.

6. After conditions have become steady, note the appearance of the colored thread and then measure the water flow rate by catching and weighing the discharge during a measured interval of time.

7. Repeat the experiment several times at rates just below, equal to, and just above the rate at which turbulence just appears.

8. Make a new series of runs in which both the flow rate and the

head loss across the tube are measured. These runs should be made at 10 or more different flow rates, covering the range from well below to well above the critical velocity. Note the appearance of the colored thread at each flow rate.

TABLE 2
Experimental Data for Viscous and Turbulent Flow of Water
Tube—10 mm. ($\frac{3}{8}$ in. I.D.)
Water temp. 23.5°C.

Run No.	Flow rate Q , lb./min.	Tank level, cm.	Manometer readings, cm.		Friction head h_f , cm. of water	Appearance of flow
			Right	Left		
1	0.59	2.2	1.9	1.7	0.2	Viscous
2	0.69	2.3	2.0	1.6	0.4	Viscous
3	1.06	2.2	1.7	1.0	0.7	Wavy
4	1.69	2.1	1.3	0.1	1.2	Slightly turbulent
5	2.06	2.0	0.7	-0.7	1.4	Fairly turbulent
6	2.82	2.1	0.2	-1.9	2.1	Turbulent
7	2.94	2.2	0.2	-2.2	2.4	Turbulent
8	3.31	2.2	-0.3	-3.2	2.9	Turbulent
9	4.31	2.1	-2.1	-9.1	7.0	Turbulent
10	5.30	2.2	-3.2	-13.2	10.0	Turbulent
11	7.07	2.2	-7.3	-24.2	16.9	Turbulent

TABLE 3
Calculated Results for Viscous and Turbulent Flow of Water in 10-mm. Tube

Run No.	Velocity of discharge V , ft./sec.	Reynolds' number, Re	Friction head H_f , ft. of water	$\frac{H_f}{V}$
1	0.298	703	0.00655	0.0220
2	0.348	923	0.0127	0.0365
3	0.534	1420	0.0222	0.0416
4	0.852	2260	0.0393	0.0461
5	1.04	2760	0.0459	0.0441
6	1.42	3760	0.0688	0.0485
7	1.48	3920	0.0787	0.0531
8	1.67	4420	0.0950	0.0569
9	2.17	5750	0.229	0.106
10	2.67	7070	0.328	0.123
11	3.56	9440	0.553	0.155

Plot a curve of flow rate Q versus head loss H_f during the progress of the experiment and take any additional data that are indicated.

9. Repeat the series of runs with at least one of the larger diameter flow tubes.

10. Measure and record the internal diameters of the flow tubes used in the experiments.

Presentation of Results. In the report of this experiment:

1. Calculate the Reynolds' number for each run.
2. Estimate the critical value of the Reynolds' number for each flow tube from the calculated values of the velocity and the observed appearance of the colored thread.
3. Plot the friction head across each tube and the ratio of friction head to velocity against the Reynolds' number on both log-log and rectangular coordinate paper, and determine the critical Reynolds' number from the breaks in the curves. Determine the slopes of the curves in both the viscous and turbulent ranges.
4. For the flow tubes used, determine whether the supposedly constant values of H_f/V in viscous flow are proportional to D^{-2} . Explain.
5. Comment upon the head loss through the entrance and the valve.

Typical Data from a Report. The experimental data for 11 runs with the 10-mm. flow tube are given in Table 2; the calculated results for these runs are given in Table 3.

Values of Reynolds' number are plotted against values of the friction head H_f and the ratio of friction head to velocity H_f/V in Figures 3 and 4. The broken lines are drawn through the low-range points with slopes that would be expected from Poiseuille's law:⁴

$$H_f = \frac{32L\mu V}{\rho g_c D^2} \quad (1)$$

where

H_f = friction loss, ft. of water.

L = length of tube, ft.

μ = viscosity, lb./(ft. \times sec.).

V = velocity, ft./sec.

ρ = density, lb./ft.³

g_c = conversion factor, = 32.174 (lb. mass \times ft.)/(lb. force \times sec.²).

D = diameter of tube, ft.

Calculations for Run No. 1

1. Velocity of discharge V , ft./sec.

$$= \frac{Q}{60 \times \rho \times A} = \frac{0.59}{60 \times 62.24 \times 5.31 \times 10^{-4}} = 0.298$$

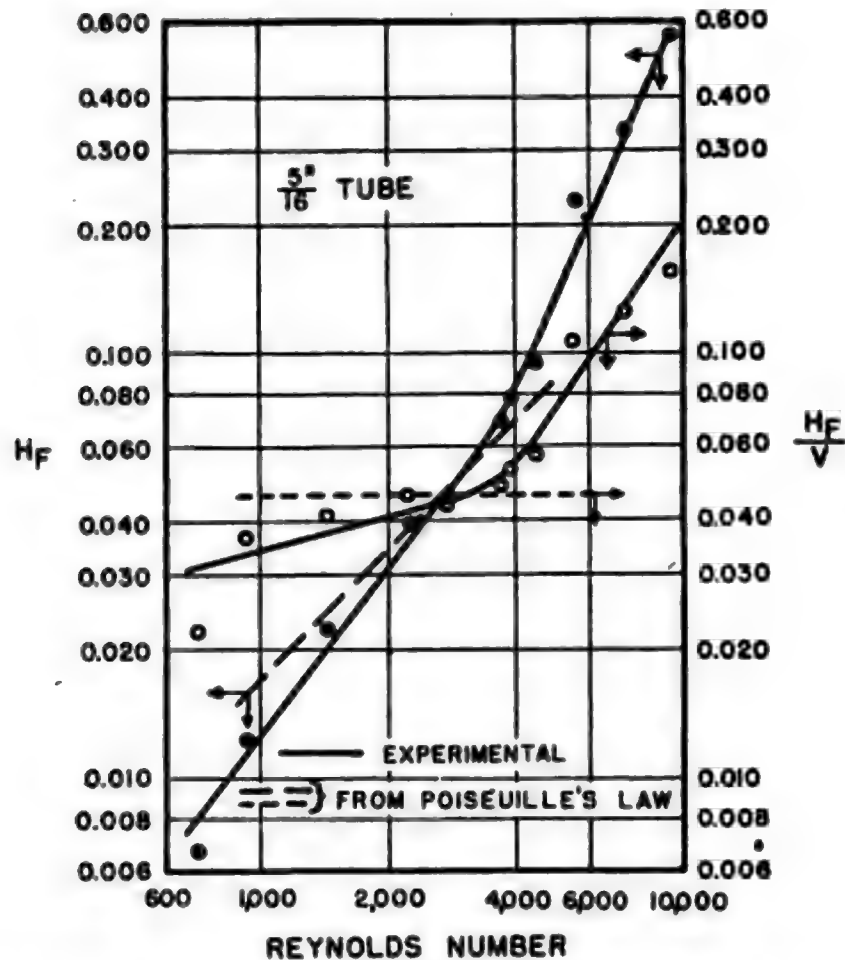


Figure 3. Effect of Velocity on Head Loss Due to Friction in a Glass Tube

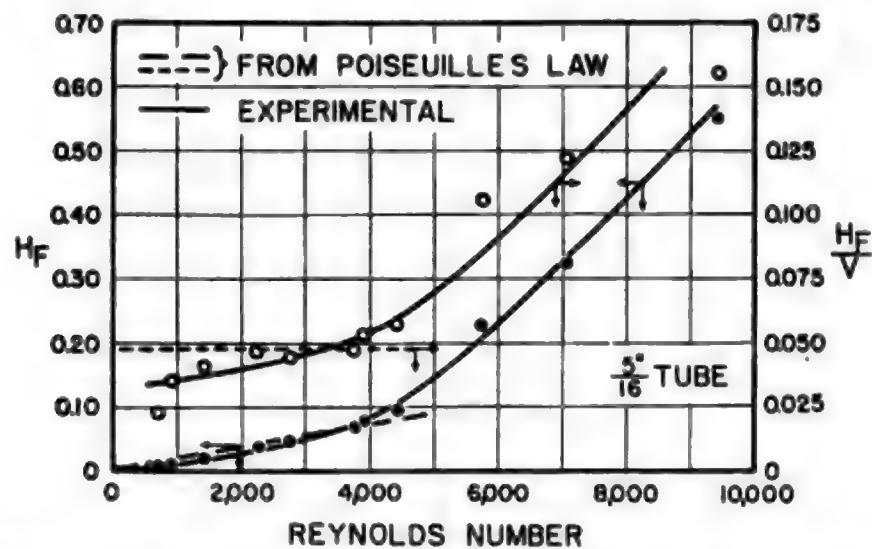


Figure 4. Effect of Velocity on Head Loss Due to Friction in a Glass Tube

where

Q = flow rate, lb./min. = 0.59.

ρ = density of water at 23.5°C., lb./ft.³ = 62.24.

A = area of the $\frac{5}{16}$ -in. I.D. flow tube, ft.²

$$= \frac{\pi(\frac{5}{16})^2}{4 \times 144} = 5.31 \times 10^{-4}.$$

2. *Reynolds' Number, Re*

$$= \frac{DV\rho}{\mu} = \frac{2.6 \times 10^{-2} \times 0.298 \times 62.24}{6.11 \times 10^{-4}} = 703$$

where

D = diameter of flow tube, ft., = $\frac{5/16}{12} = 2.60 \times 10^{-2}$.

V = velocity of discharge, ft./sec. = 0.298.

ρ = density of water at 23.5°C., lb./ft.³ = 62.24.

μ = viscosity of water at 23.5°C. = 0.910 centipoises

$$= 0.910 \times 0.000672 = 6.11 \times 10^{-4} \text{ lb.}/(\text{ft.} \times \text{sec.}).$$

3. *Friction Head H_f , ft. of water*

$$= \frac{h_f}{2.54 \times 12} = \frac{0.2}{30.5} = 0.00655$$

where h_f is the friction head in cm. of water.

Comments. 1. The critical region, as determined by visual observation, extended between Runs No. 3 and 5, which corresponds to a range in the Reynolds' number of about 1400 to 2800; this is in substantial agreement with most of the values found in the literature.

2. According to Figures 3 and 4, the critical region as determined from the pressure drop data seems to be somewhat higher.

NOMENCLATURE

A = cross-sectional area of flow tube, ft.²

D = inside diameter of flow tube, ft.

g_c = conversion factor = 32.174 (lb. mass \times ft.)/(lb. force \times sec.²).

H_f = friction loss, ft. of water.

h_f = friction loss, cm. of water.

L = length of flow tube, ft.

Q = rate of flow, lb./min.

μ = viscosity, lb./(ft. \times sec.).

V = velocity, ft./sec.

ρ = density, lb./ft.³

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A FLUID-FLOW APPARATUS

Designed and Constructed by

The Department of Chemical Engineering
Drexel Institute of Technology*

Description. This apparatus, shown in Figure 1, is constructed of several lengths of 2-inch standard pipe, one length of 1½-inch pipe, and a number of different fittings; the system also includes a Venturi meter. Open-end glass tubes are attached to fourteen piezometer openings to permit the measurement of pressure heads at various points in the system.

Materials and Cost. The materials used in constructing this apparatus are given in Table 1. The cost of these items is approximately \$120.

The following equipment, which is not included in the bill of materials, is needed when conducting an experimental run with this assembly:

1. Platform scale of 1000-pound capacity.
2. Weighing tank of 100-gallon capacity.
3. Stop watch.
4. Measuring tape.
5. Thermometer, graduated in 1°C.

Notes on Construction. 1. After the required lengths of pipe have been cut and threaded, the ends should be reamed to remove burrs.

2. The piezometer openings are made by drilling ¼-inch holes and filing the openings on the inside of the pipe to remove burrs.** One-eighth inch short nipples are then welded over each opening.

3. Connections between the nipples and the glass tubes are preferably made with ⅜-inch copper tubing. Compression fittings can be used for attaching the copper tubing to the nipples, and rubber tubing is satisfactory for the copper tubing-glass connections.

Comments on Design. 1. The dimensions of this unit and the number and types of fittings can be modified to conform to the space available and individual preference.

2. The height of the supply tank is largely determined by the head

* Reported by H. T. Ward.

** A half-round file attached to a length of wood or small diameter pipe is very convenient for removing burrs.



room available; it is preferable to have a greater available head than that indicated here.

3. The use of a pump to return the discharge to the supply tank is desirable when the water supply is limited and is necessary when fluids other than water are used.

4. A mercury draft gage can be advantageously used instead of the open-end piezometer tubes for the Venturi meter, particularly when the head room is limited.

TABLE 1
Bill of Materials for Fluid Flow Apparatus
Drexel Institute of Technology

Quantity	Description	Size	Material
91 ft.	Std. pipe	2 in.	Black wrought steel
20 ft.	" "	1½ in.	" " "
1 ft.	" "	½ in.	" " "
14	Std. nipple	½ in. x 4 in. long	" " "
1	" "	1 in. x 4 in. long	" " "
3	Std. 90° elbow	2 in.	Black malleable iron
1	Long sweep 90° elbow	2 in.	" " "
1	Special wide pattern return bend; 6 in. center to center	2 in.	" " "
2	Std. reducer	2 x 1½ in.	" " "
1	Std. coupling	2 in.	" " "
3	Std. gate valve	2 in.	Brass
1	Std. globe valve	½ in.	"
1	Venturi meter	2 in. (1½ in. throat)	Cast iron
14	Piezometer tube	10 mm. x 5 ft. long	Glass
1	Supply tank	50 gal.	Steel

Operation. Water is drawn from overhead supply tank *B* and is passed through the system at various rates of flow, which are regulated by adjusting control valve *F*.

The discharge at each velocity is collected in tank *G* and weighed over a measured interval of time to permit the calculation of the velocity.

The height of the water in each of the glass tubes is measured during each run. The initial, or zero, heights are checked before and after each series of runs by filling the system completely with water and closing valve *F*.

TYPICAL EXPERIMENT

- Objects.** 1. To determine the friction loss in pipe and fittings.
2. To calibrate the Venturi meter.

Procedure. 1. Study the design and construction of the apparatus to become thoroughly familiar with the method of operation.

2. Measure the distances between the piezometer stations.
3. Measure the zero heights of water in the piezometer tubes with the system completely filled with water but with no water flowing.
4. Make five runs within the limits of the Venturi piezometer tubes and then close the Venturi piezometer connections and make two or three additional runs to the limit of the last piezometer tube. During each run, maintain a constant head in tank *B* by adjusting valve *A* to provide a slight but continuous discharge through drain *C*.
5. Take at least two sets of piezometer readings for each run.
6. Weigh the discharge over a 3 to 4-minute interval for each run.
7. Measure the temperature of the water several times during the experiment.
8. From the data obtained, calculate for each rate of flow:
 - a. The velocities in the 1½- and 2-inch pipes.
 - b. The head loss, in feet of water, across:
 - (1) Ten feet of 2-inch pipe.
 - (2) Ten feet of 1½-inch pipe.
 - (3) The Venturi meter.
 - (4) The 2-inch elbow.
 - (5) The 2-inch gate valve.
 - (6) The pipe contraction.
 - (7) The pipe enlargement.
9. Plot:
 - a. Curves showing piezometer readings in feet of water versus the distance in feet to piezometer stations.
 - b. Calibration curves for the Venturi meter on both rectangular coordinate paper and log-log paper in terms of pressure drop in feet of water versus flow in cubic feet per second.
10. From the log-log plot develop an equation that applies to the Venturi meter, and calculate the value of the Venturi constant.
11. Compare:
 - a. The experimentally determined head losses in the pipe and fittings with those obtained by using the pipe friction curve and data given in Badger and McCabe.
 - b. The experimentally determined Venturi constant with the generally accepted value.

TABLE 2
Measurements of Fluid Flow Apparatus

Piezometer stations	Fitting and length of pipe
1-2	Gate valve plus 10 ft. of 2-in. pipe
2-3	Std. coupling plus 5.1 ft. of 2-in. pipe
3-6	Venturi meter plus 22.7 ft. of 2-in. pipe
6-7	Elbow plus 7.5 ft. of 2-in. pipe
7-8	Return bend plus 5 ft. of 2-in. pipe
8-9	Long sweep elbow plus 7.3 ft. of 2-in. pipe
9-10	4.95 ft. of 2-in. pipe
10-11	Reducer plus 2.6 ft. of 2-in. pipe plus 2.6 ft. of 1½-in. pipe
11-12	14.9 ft. of 1½-in. pipe
12-13	Reducer plus 2.6 ft. of 1½-in. pipe plus 2.6 ft. of 2-in. pipe
13-14	10 ft. of 2-in. pipe

TABLE 3
Experimental Data on the Friction Loss in Pipe and Fittings
(Average water temperature = 68°F.)

Run No.	Time of run, min:sec	Wt. of discharge, lb.	Head in tank, ft.	Piezometer Station													
				1	2	3	4	5	6	7	8	9	10	11	12	13	14
				Head in feet of water													
1	3:53	279	4.47	4.44	4.42	4.40	4.35	3.39	4.10	4.08	4.06	4.02	4.01	3.95	3.89	3.90	3.88
				4.43	4.42	4.41	4.34	3.39	4.11	4.08	4.06	4.01	4.00	3.95	3.89	3.90	3.88
2	4:13	413	4.48	4.40	4.35	4.33	4.28	2.49	3.88	3.85	3.79	3.73	3.72	3.62	3.43	3.44	3.41
				4.41	4.35	4.34	4.27	2.49	3.87	3.85	3.79	3.72	3.74	3.62	3.43	3.44	3.42
3	4:11	500	4.48	4.35	4.29	4.26	4.17	1.52	3.59	3.52	3.44	3.37	3.36	3.21	2.93	2.95	2.90
				4.35	4.29	4.27	4.16	1.50	3.59	3.52	3.44	3.38	3.37	3.20	2.93	2.96	2.90
4	4:15	550	4.48	4.32	4.25	4.21	4.13	1.07	3.46	3.37	3.28	3.19	3.18	3.02	2.70	2.73	2.66
				4.33	4.26	4.20	4.14	1.07	3.47	3.38	3.28	3.19	3.18	3.02	2.71	2.73	2.66
5	3:53	548	4.50	4.30	4.23	4.19	4.09	0.47	3.26	3.16	3.08	2.97	2.95	2.74	2.37	2.40	2.33
				4.31	4.23	4.19	4.09	0.46	3.26	3.16	3.08	2.98	2.95	2.74	2.38	2.40	2.33
6	3:47	596	4.50	4.28	4.18	4.13	4.00	—	2.95	2.82	2.73	2.62	2.60	2.34	1.86	1.90	1.82
				4.28	4.20	4.12	4.00	—	2.95	2.82	2.74	2.61	2.60	2.35	1.87	1.90	1.83
7	3:25	592	4.50	4.22	4.09	4.03	3.90	—	2.60	2.43	2.33	2.18	2.14	1.84	1.27	1.30	1.20
				4.21	4.09	4.04	3.91	—	2.61	2.43	2.34	2.18	2.14	1.84	1.27	1.31	1.21
8	3:05	590	4.50	4.17	4.03	3.95	3.80	—	2.20	2.02	1.88	1.72	1.69	1.32	0.64	0.68	0.56
				4.16	4.03	3.95	3.81	—	2.21	2.02	1.88	1.72	1.69	1.33	0.64	0.69	0.56

Typical Data. The distances between various piezometer stations were measured, and then eight experimental runs were made at different rates of flow. The measurements are given in Table 2, the experimen-

TABLE 4
Loss in Head Between Piezometer Stations
(Average values for two readings)

Stations	Run number							
	1	2	3	4	5	6	7	8
	Feet of water							
1-2	0.015	0.055	0.06	0.07	0.075	0.09	0.125	0.135
2-3	0.015	0.015	0.025	0.05	0.04	0.065	0.055	0.08
3-4	0.30	0.45	0.675	0.74	0.93	1.175	1.43	1.745
4-5	0.955	1.785	2.655	3.065	3.625	—	—	—
6-7	0.025	0.025	0.07	0.09	0.10	0.13	0.175	0.185
7-8	0.02	0.06	0.08	0.09	0.08	0.085	0.095	0.14
8-9	0.045	0.065	0.065	0.095	0.105	0.12	0.155	0.16
9-10	0.01	-0.005	0.01	0.01	0.025	0.015	0.04	0.03
10-11	0.055	0.11	0.16	0.16	0.21	0.255	0.30	0.365
11-12	0.06	0.19	0.275	0.315	0.365	0.48	0.57	0.685
12-13	-0.01	-0.01	-0.025	-0.025	-0.025	-0.035	-0.035	-0.045
13-14	0.02	0.025	0.055	0.07	0.07	0.075	0.10	0.125

TABLE 5
Calculated Flow Rates and Velocities

Sym- bol	Item	Run number							
		1	2	3	4	5	6	7	8
q_m	Rate of flow, ft. ³ /min.	1.16	1.57	1.92	2.08	2.26	2.53	2.78	3.08
q_v	Rate of flow, ft. ³ /sec.	0.0193	0.0262	0.0320	0.0346	0.0377	0.0421	0.0465	0.0512
u_1	Vel. in 2-in. pipe, ft./sec.	0.828	1.125	1.375	1.486	1.620	1.810	1.998	2.200
u_2	Vel. in 1½-in. pipe, ft./sec.	1.366	1.854	2.263	2.448	2.668	2.980	3.290	3.622

tal data for the eight runs are given in Table 3, and the calculated results for these runs are given in Tables 4 to 7. To illustrate the drop in pressure through the apparatus, the pressure heads at various piezometer stations for Run No. 1 are plotted against distance in feet in Figure 2.

TABLE 6
Loss in Head Calculated from Experimental Data

Symbol	Loss in:	Run number							
		1	2	3	4	5	6	7	8
		Feet of water							
F_1	10 ft. of 2-in. pipe	0.02	0.025	0.05	0.07	0.07	0.075	0.10	0.125
F_2	10 ft. of 1½-in. pipe	0.04	0.13	0.18	0.21	0.25	0.31	0.38	0.46
F_3	Venturi meter	0.255	0.403	0.550	0.581	0.771	1.005	1.203	1.461
F_4	2-in. elbow	0.010	0.006	0.029	0.037	0.047	0.074	0.100	0.091
F_5	2-in. gate valve	-0.005	0.020	0.005	0.000	0.005	0.015	0.025	0.010
F_6	2 to 1½ in. contraction	0.021	0.037	0.046	0.028	0.058	0.067	0.069	0.083
F_7	1½ to 2-in. enlargement	-0.008	-0.016	-0.035	-0.040	-0.037	-0.049	-0.054	-0.064

TABLE 7
Loss in Head Calculated from Pipe Friction Equation and from Equivalent Lengths of Standard Fittings

Sym- bol	Item	Run number							
		1	2	3	4	5	6	7	8
		Feet of water							
Re_1	Reynolds' No., 2-in. pipe	13,210	17,960	21,970	23,730	25,880	28,900	31,900	35,150
Re_2	Reynolds' No., 1½-in. pipe	17,000	23,060	28,160	30,450	33,180	37,090	40,920	45,080
	Loss in:	Feet of water							
F_8	10 ft. of 2-in. pipe	0.020	0.036	0.046	0.057	0.066	0.082	0.096	0.115
F_9	10 ft. of 1½-in. pipe	0.067	0.115	0.164	0.190	0.221	0.271	0.323	0.386
F_{10}	Venturi meter	0.115	0.214	0.319	0.368	0.435	—	—	—
F_{11}	2 in. elbow	0.007	0.013	0.016	0.020	0.028	0.029	0.034	0.041
F_{12}	2-in. gate valve	0.003	0.004	0.005	0.007	0.008	0.009	0.011	0.013
F_{13}	2 to 1½-in. contraction	0.007	0.013	0.020	0.023	0.028	0.034	0.042	0.051
F_{14}	1½ to 2-in. enlargement	0.005	0.008	0.012	0.014	0.017	0.021	0.026	0.031

The calibration data for the Venturi meter are plotted on a rectangular coordinate scale in Figure 3, and on a log-log scale in Figure 4.

The log-log plot gives a straight line whose slope is 0.5 and whose intercept is 0.0197; therefore, the equation of the line is

$$Q = 0.0197(\Delta H)^{0.5} \quad (1)$$

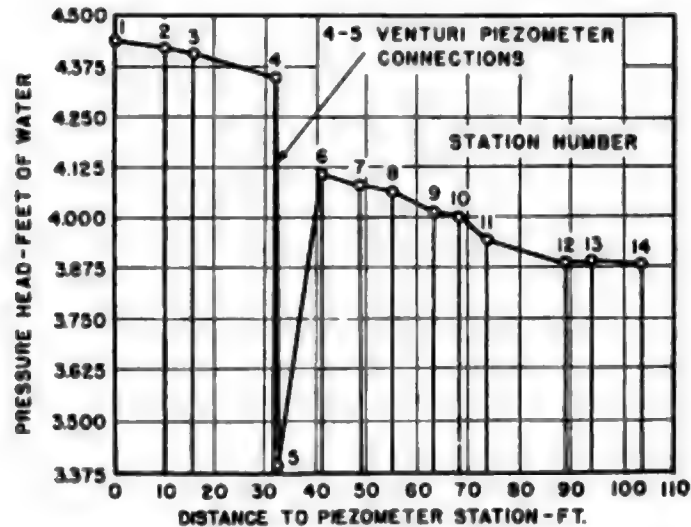


Figure 2. Pressure Heads at Piezometer Stations for Run No. 1

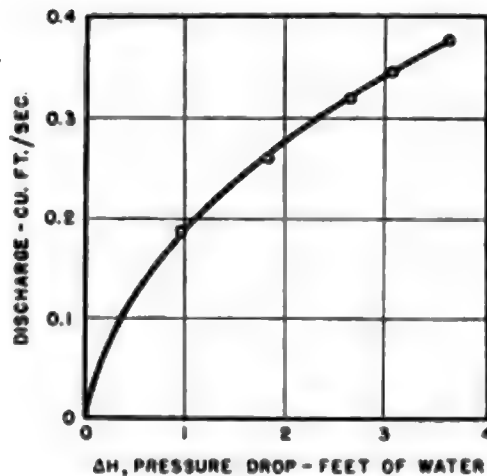


Fig. 3

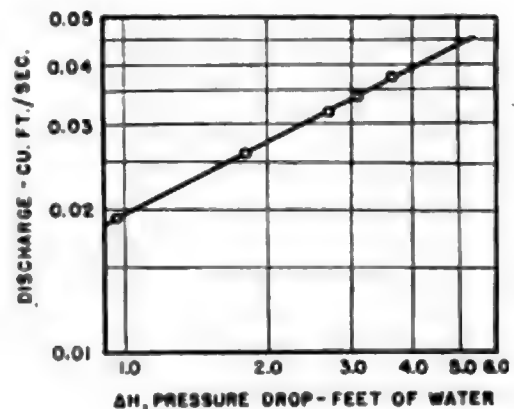


Fig. 4

Figure 3. Rectangular Coordinate Plot of Venturi Meter Calibration

Figure 4. Log-log Plot of Venturi Meter Calibration

where

Q = discharge in ft.³/sec.

ΔH = pressure drop in ft. of water.

The cross-sectional area of the Venturi throat is 0.00258 ft.²; therefore

$$u = \frac{0.0197}{0.00258} \sqrt{\Delta H} = 7.7 \sqrt{\Delta H} \quad (2)$$

where

$$\begin{aligned} u &= \text{velocity in ft./sec.} \\ &= Q/\text{area} = Q/0.00258. \end{aligned}$$

The Venturi equation is

$$u = C_v \sqrt{2g_c \Delta H} \quad (3)$$

Therefore,

$$C_v \sqrt{2g_c \Delta H} = 7.7 \sqrt{\Delta H}$$

and

$$C_v = \frac{7.7}{\sqrt{2 \times 32.2}} = 0.96$$

where

C_v = Venturi constant.

g_c = conversion factor, (lb. mass \times ft.)/(lb. force \times sec.²).

This value for the Venturi constant compares favorably with the generally accepted value¹ of 0.98.

SAMPLE CALCULATIONS FOR RUN NO. 1

1. Head Losses Calculated from Experimental Data*.

q_m = flow rate, ft.³/min.

$$= (279/3.86) \times 1/62.32 = 1.16$$

where

279 = weight of discharge in 3.86 min.

62.32 = density of water at 68°F.

q_v = flow rate, ft.³/sec.

$$= 1.16/60 = 0.0193$$

u_1 = velocity in 2-in. pipe, ft./sec.

$$= (0.0193/3.355) \times 144 = 0.828$$

where

3.355 = cross-sectional area of 2-in. pipe, in.²

u_2 = velocity in 1-in. pipe, ft./sec.

$$= (0.0193/2.036) \times 144 = 1.366$$

where

2.036 = cross-sectional area of 1½-in. pipe, in.²

F_1 = head loss in 10 ft. of 2-in. pipe

$$= 0.02 \text{ (Stations 13-14)}$$

* All head losses are expressed in feet of water.

$$F_2 = \text{head loss in 10 ft. of } 1\frac{1}{2}\text{-in. pipe}$$

$$= 0.06 \times 10/14.9 = 0.04$$

where

$$0.06 = \text{head loss in 14.9 ft. of pipe (Stations 11-12).}$$

$$F_3 = \text{head loss in Venturi meter}$$

$$= 0.30 - F_1 \times 22.7/10 = 0.30 - 0.02 \times 22.7/10 = 0.255$$

where

$$0.30 = \text{head loss in Venturi meter plus 22.7 ft. of 2-in. pipe, (Stations 3-6).}$$

$$F_4 = \text{head loss in 2-in. elbow}$$

$$= 0.025 - F_1 \times 7.5/10 = 0.025 - 0.02 \times 7.5/10 = 0.010$$

where

$$0.025 = \text{head loss in elbow plus 7.5 ft. of pipe (Stations 6-7).}$$

$$F_5 = \text{head loss in 2-in. standard gate valve}$$

$$= 0.015 - F_1 = 0.015 - 0.02 = -0.005$$

where

$$0.015 = \text{head loss in gate valve plus 10 ft. of pipe (Stations 1-2).}$$

$$F_6 = \text{head loss in 2 to } 1\frac{1}{2}\text{-in. pipe contraction}$$

$$= 0.055 - (F_1 \times 2.6/10) - (F_2 \times 2.6/10) - \left(\frac{u_2^2 - u_1^2}{2g_c} \right)$$

$$= 0.055 - (0.02 \times 2.6/10) - (0.04 \times 2.6/10) - \left(\frac{1.366^2 - 0.828^2}{2 \times 32.2} \right)$$

$$= 0.021.$$

where

$$0.055 = \text{head loss in contraction plus 2.6 ft. of 2-in. pipe plus 2.6 ft. of } 1\frac{1}{2}\text{-in. pipe. (Stations 10-11).}$$

$$F_7 = \text{head loss in } 1\frac{1}{2} \text{ to 2-in. enlargement}$$

$$= -0.01 - (F_1 \times 2.6/10) - (F_2 \times 2.6/10) + \left(\frac{u_2^2 - u_1^2}{2g_c} \right)$$

$$= -0.01 - (0.02 \times 2.6/10) - (0.04 \times 2.6/10)$$

$$+ \left(\frac{1.366^2 - 0.828^2}{2 \times 32.2} \right)$$

$$= -0.008$$

where

$$-0.01 = \text{head loss in enlargement plus 2.6 ft. of 2-in. pipe plus 2.6 ft. of } 1\frac{1}{2}\text{-in. pipe (Stations 12-13).}$$

2. Head Losses Calculated from Equations and Values from Literature.*Re₁* = Reynolds' number for 2-in. pipe

$$= \frac{D_1 u_1 \rho}{\mu} = \frac{(2.067/12) \times 0.828 \times 62.32}{1.000 \times 0.000672} = 13,210$$

Re₂ = Reynolds' number for 1½-in. pipe

$$= \frac{D_2 u_2 \rho}{\mu} = \frac{(1.610/12) \times 1.366 \times 62.32}{1.000 \times 0.000672} = 17,000$$

F₈ = head loss in 10 ft. of 2-in. pipe

$$= \frac{u_1^2 L}{D_1} \phi(Re_1) = \frac{(0.828)^2 \times 10}{2.067/12} \times 0.00047 = 0.020$$

where

0.00047 = friction factor; from Figure 14, Badger and McCabe².*F₉* = head loss in 10 ft. of 1½-in. pipe

$$= \frac{u_2^2 L}{D_2} \phi(Re_2) = \frac{(1.366)^2 \times 10}{(1.610/12)} \times 0.00049 = 0.067$$

F₁₀ = permanent head loss in Venturi meter

$$= 0.955 \times 1/9 = 0.106$$

where

0.955 = head loss in Venturi meter (Stations 4-5).

1/9 = average of generally accepted value of 1/8 to 1/10 for permanent head loss in Venturi.¹*F₁₁* = head loss in 2-in. elbow

$$= (0.02/10) \times 3.55 = 0.007$$

where

0.02 = head loss for 10 ft. of 2-in. pipe = *F₁*.3.55 = feet of pipe equivalent to a 2-in. elbow.³*F₁₂* = head loss in 2-in. gate valve

$$= (0.02/10) \times 1.32 = 0.003$$

where

1.32 = ft. of pipe equivalent to a 2-in. gate valve.³*F₁₃* = head loss in 2 to 1½-in. contraction⁴

$$= K \frac{u_1^2}{2g_s} = \frac{0.25 \times (1.366)^2}{2 \times 32.2} = 0.007$$

where

0.25 = contraction coefficient for a ratio of areas of $\frac{2.306}{3.355}$.

$$F_{14} = \text{head loss in } 1\frac{1}{2} \text{ to 2-in. enlargement}^5$$

$$= \frac{(u_2 - u_1)^2}{2g_c} = \frac{(1.366 - 0.828)^2}{2 \times 32.2} = 0.005$$

NOMENCLATURE

- D_1 = inside diameter of 2-in. pipe, ft.
 D_2 = inside diameter of 1-in. pipe, ft.
 F = head loss due to friction, ft. of fluid.
 K = contraction coefficient.
 L = length of pipe, ft.
 Re = Reynolds' number.
 u_1 = average linear velocity in 2-in. pipe, ft./sec.
 u_2 = average linear velocity in 1-in. pipe, ft./sec.
 μ = viscosity of water, lb./(ft. \times sec.).
 ρ = density of water, lb./ft.³

REFERENCES

1. BADGER, AND McCABE, "Elements of Chemical Engineering," McGraw-Hill Book Company, New York, 2nd ed., 1936, p. 48.
2. Ibid., p. 37.
3. Ibid., p. 631.
4. Ibid., p. 40.
5. Ibid., p. 39.

A FLUID-FLOW APPARATUS

Designed and Constructed by
The Department of Chemical Engineering
University of Florida*

Description. This apparatus, shown in Figure 1, is made of standard 1- and $\frac{1}{2}$ -inch galvanized steel pipe and fittings and a Venturi meter of home construction. Piezometer openings are connected to a mercury manometer to permit the measurement of the pressure drops across various sections of the apparatus. A centrifugal pump is used to circulate the fluid from a tank through any desired part of the system. The apparatus is designed to occupy a minimum of floor space; one portion is placed against a wall and the remainder is suspended from the ceiling at right angles to the wall.

Materials and Cost. The materials for this fluid-flow apparatus cost approximately \$160; these materials are listed in Table 1.

In addition to the fluid-flow apparatus itself, the following additional equipment is needed for experimental work:

1. Platform scale or other suitable weighing equipment: capacity approximately 50 pounds.
2. Weighing bucket: capacity approximately 5 gallons.
3. Stop watch.

Notes on Construction. *The piezometer openings* are made by threading $\frac{1}{2}$ -inch brass nipples into tapped holes and then soldering the nipples to the pipe. The nipples are threaded just sufficiently to hold them in place while they are being soldered.

The various piezometer openings are connected to a single mercury manometer through $\frac{1}{2}$ -inch brass cocks and $\frac{1}{2}$ -inch copper tubing.

The Venturi meter is made by pouring molten lead around two steel forms held closely in a 1-inch brass pipe. The two forms, 24 inches in total length and $\frac{1}{4}$ -inch at the throat, are machined so that the small ends form a ball-and-socket joint. The larger ends of each form are tapped to receive bolts which assist in the removal of the forms after the lead has solidified. The lead is poured through a $\frac{1}{2}$ -inch opening at the throat; and this opening later serves as one of the manometer taps. The other $\frac{1}{2}$ -inch tap is installed $1\frac{1}{2}$ inch upstream from the point where the contraction starts. The Venturi meter is fitted into the line by means of couplings.

* Reported by W. H. Beisler.

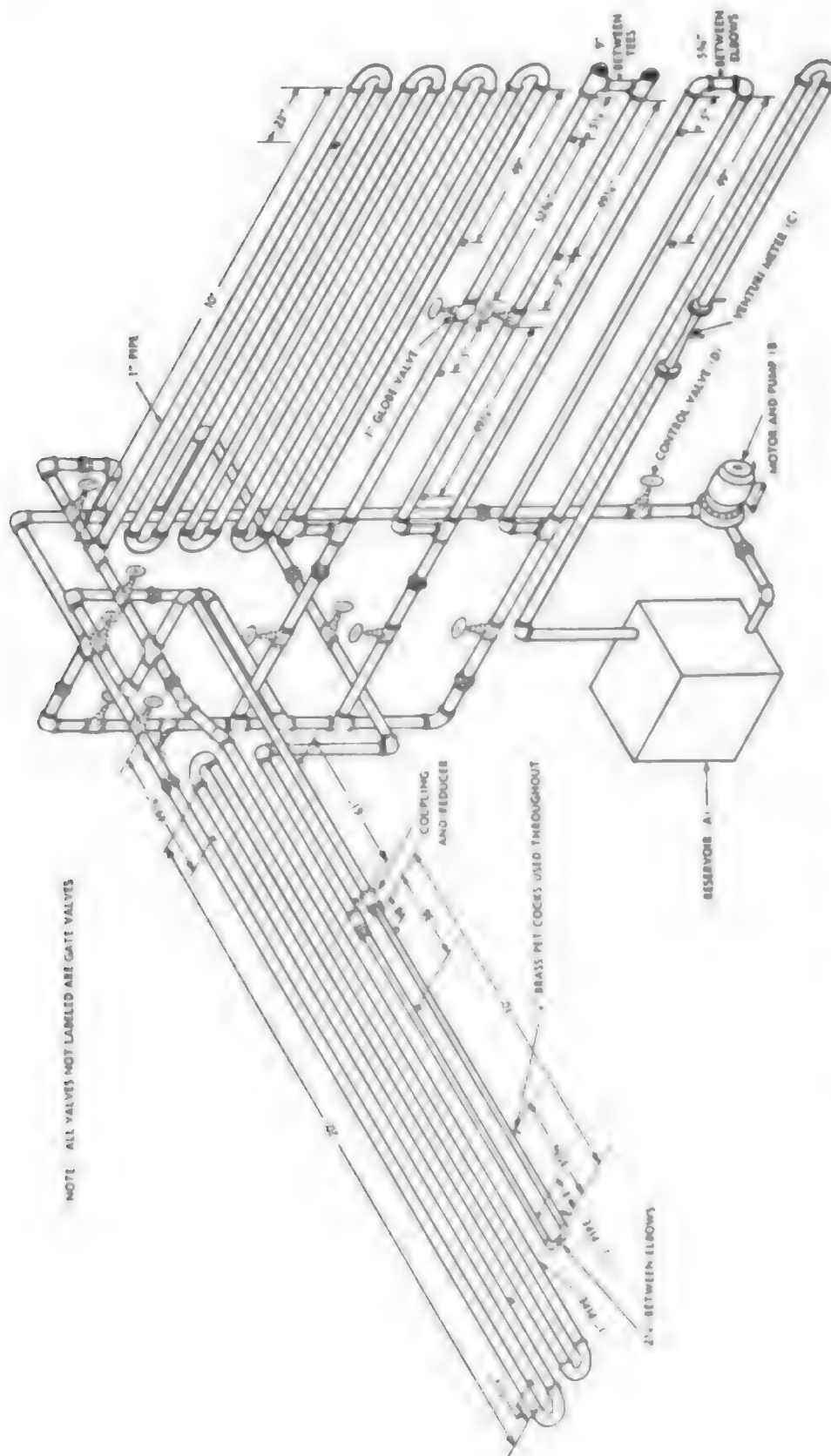


Figure 1. Fluid-Flow Apparatus—University of Florida

Comments on Design. This apparatus is designed so that the fluid does not have to pass through the entire system at any one time, but can be sent through individual parts; this permits the attainment of greater velocities.

Comments on the Choice of Fluid. The choice of fluid is one of individual preference. A mixture of light lubricating oil and kerosene

TABLE 1
Bill of Materials for Fluid-Flow Apparatus
University of Florida

Quantity	Description	Size	Material
260 ft.	Std. pipe	1 in.	Galv. wrought steel
20 ft.	" "	$\frac{1}{2}$ in.	" " "
14	Std. tee	1 in.	Galv. malleable iron
12	Std. 90° elbow	1 in.	" " "
2	" " "	$\frac{1}{2}$ in.	" " "
5	" 45° "	1 in.	" " "
13	Ground-joint union	1 in.	" " "
2	Bushing	1 x $\frac{1}{2}$ in.	" cast iron
7	Std. coupling	1 in.	" malleable iron
1	Std. cross	1 in.	" " "
3	Plug	1 in.	Galv. cast iron
11	Std. close-pattern return bend	1 in.	Galv. malleable iron
12	Low-pressure gate valve	1 in.	Brass
1	Std. globe valve	1 in.	"
20	Pet cock	$\frac{1}{2}$ in.	"
1	Venturi meter	1 in. ($\frac{1}{8}$ in. throat)	Brass and lead
100 ft.	Tubing	$\frac{1}{2}$ in.	Copper
14	Tee	$\frac{1}{2}$ in.	Brass
4	Union	$\frac{1}{2}$ in.	"
1	Reservoir	23 $\frac{1}{2}$ x 23 $\frac{1}{2}$ x 23 in.	Steel
1	Centrifugal pump and motor	1 H.P.	—

having a viscosity of 4.5 centipoises and a specific gravity of 0.81 has been found to give satisfactory results. The use of such an oil mixture has the advantage of being non-corrosive, and its viscosity is such that the change in flow from the viscous to the turbulent region occurs at approximately the middle of the velocity range as obtained with a 1-H.P. centrifugal pump. This makes it possible to determine the head loss for both viscous and turbulent flow.

Operation. Fluid, from reservoir *A* is forced by pump *B* through any desired part of the system, and returns to the reservoir after passing

TABLE 2
Calibration Data of the Venturi Meter

Venturi reading, in. Hg	Weight of oil, Kg.	Time, sec.	Volume of oil, ft. ³	Flow rate, ft. ³ /sec.	Vel. in 1-in. pipe, ft./sec.	Vel. in $\frac{1}{2}$ -in. pipe, ft./sec.
42.25	12.647	17.2	0.551	0.0320	5.33	15.20
38.50	11.711	16.8	0.511	0.0304	5.06	14.40
36.63	11.895	17.6	0.519	0.0295	4.92	13.96
34.38	12.730	19.1	0.555	0.0291	4.85	13.78
30.75	12.691	20.0	0.553	0.0276	4.61	13.12
29.63	12.301	20.0	0.536	0.0268	4.47	12.70
22.50	12.064	22.4	0.525	0.0234	3.91	11.11
17.13	12.221	26.0	0.532	0.0204	3.41	9.70
15.25	12.610	28.1	0.550	0.0196	3.27	9.29
12.19	12.814	32.2	0.559	0.0174	2.90	8.23
9.75	12.889	36.1	0.562	0.0156	2.60	7.39
6.88	12.980	42.6	0.506	0.0133	2.22	6.30
4.25	13.050	52.8	0.569	0.0108	1.80	5.11
1.56	12.579	83.1	0.548	0.0066	1.10	3.13

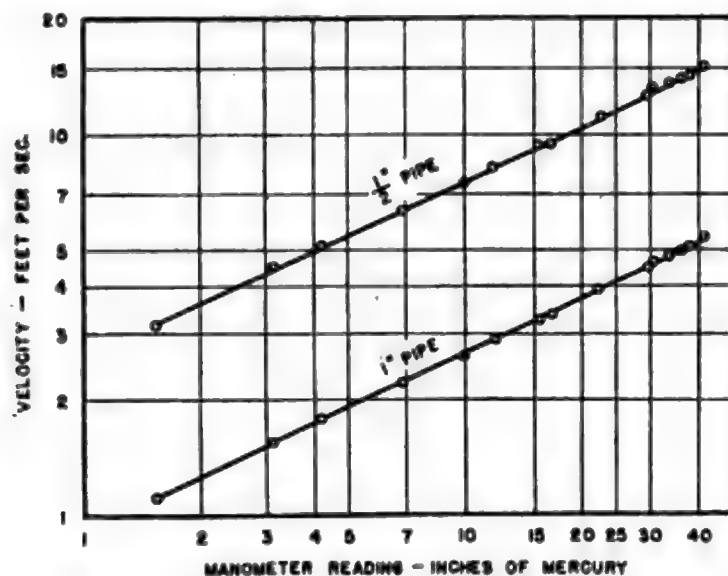


Figure 2. Calibration Curves for Venturi Meter

through Venturi meter *C*. The rate of flow is controlled by adjusting valve *D*.

The Venturi meter is first calibrated, and is later used in measuring the rates of flow during the friction-loss experiments. The calibration

is performed by collecting and weighing the discharge over a measured interval of time.

TYPICAL EXPERIMENT

Object. To determine the head loss due to friction in 1- and $\frac{1}{2}$ -inch galvanized pipe and across various fittings.

TABLE 3
Loss of Head in 1-inch Pipe and Fittings

Venturi reading, in. Hg	Vel. in 1-in. pipe, ft./sec.	Head loss, in. Hg	Head loss, ft. of oil	Venturi reading, in. Hg	Vel. in 1-in. pipe, ft./sec.	Head loss, in. Hg	Head loss, ft. of oil
(A) Three close return bends plus 72.28 ft. of pipe				(D) Close return bend plus 4.58 ft. of pipe			
38.50	5.05	10.63	14.00	38.50	5.05	1.10	1.45
34.00	4.78	9.63	12.65	34.00	4.78	0.99	1.31
26.88	4.28	7.75	10.15	26.88	4.28	0.78	1.03
18.25	3.56	5.50	7.23	18.25	3.56	0.55	0.72
11.65	2.87	3.70	4.86	11.65	2.87	0.38	0.50
7.90	2.39	2.65	3.48	7.90	2.39	0.28	0.37
4.35	1.79	1.66	2.18	4.35	1.79	0.18	0.24
1.90	1.20	0.86	1.13	1.90	1.20	0.10	0.13
(B) Gate valve plus 4.83 ft. of pipe				(E) Two tees plus 5.28 ft. of pipe			
38.75	5.09	0.70	0.92	38.63	5.06	1.45	1.92
31.13	4.60	0.60	0.79	30.25	4.52	1.20	1.58
21.00	3.80	0.41	0.54	20.00	3.72	0.85	1.12
11.90	2.90	0.28	0.34	11.50	2.85	0.55	0.72
8.00	2.40	0.20	0.26	7.38	2.31	0.36	0.47
4.10	1.74	0.12	0.16	3.81	1.69	0.22	0.29
2.10	1.26	0.08	0.11	2.00	1.23	0.16	0.21
(C) Globe valve plus 4.80 ft. of pipe				(F) Two elbows plus 4.95 ft. of pipe			
38.38	5.04	2.50	3.29	38.00	5.03	1.10	1.45
28.63	4.41	1.95	2.56	29.75	4.48	0.93	1.22
17.88	3.52	1.23	1.62	18.63	3.60	0.61	0.80
10.63	2.75	0.75	0.99	11.63	2.89	0.41	0.54
7.75	2.36	0.60	0.79	6.70	2.21	0.28	0.37
2.85	1.46	0.27	0.35	2.60	1.39	0.17	0.22

Procedure. 1. Determine the viscosity and specific gravity of the fluid, using an Ostwald viscosimeter, and a pycnometer or specific gravity balance.

2. Calibrate the Venturi meter, and plot velocity in feet per second against manometer deflection in inches of mercury.

TABLE 4

Loss of Head in $\frac{1}{4}$ -inch Pipe and Fittings and in Contraction and Expansion

Venturi reading, in. Hg	Vel. in $\frac{1}{4}$ -in. pipe, ft./sec.	Head loss, in. Hg	Head loss, ft. of oil	Venturi reading, in. Hg	Vel. in $\frac{1}{4}$ -in. pipe, ft./sec.	Head loss, in. Hg	Head loss, ft. of oil
(A) 16.33 ft. of $\frac{1}{4}$ in.-pipe plus 2 elbows				(B) 3.83 ft. of $\frac{1}{4}$ -in. pipe plus 2 elbows			
32.67	13.3	33.38	43.8	33.68	13.6	14.69	19.3
30.50	13.0	31.75	41.7	30.25	12.9	13.44	17.7
29.00	12.7	30.25	39.8	27.75	12.4	12.38	16.3
26.50	12.1	27.88	36.7	24.25	11.6	10.94	14.4
23.88	11.6	25.25	33.2	20.88	10.8	9.50	12.5
21.13	10.9	22.75	29.9	18.00	10.1	8.31	10.9
16.13	9.6	17.88	23.5	15.50	9.4	7.31	9.6
12.00	8.3	13.88	18.3	12.88	8.6	6.19	8.14
8.50	7.0	10.44	13.7	10.00	7.6	5.00	6.57
6.63	6.3	8.38	11.0	7.38	6.6	3.81	5.01
4.50	5.2	6.38	8.38	5.12	5.5	2.85	3.75
2.88	4.2	4.63	6.08	2.45	3.9	1.70	2.23
0.90	2.4	1.52	2.00	0.78	2.26	1.05	1.38
1.75	3.3	3.10	4.08	0.35	1.54	0.67	0.88
3.10	4.4	4.70	6.18	1.26	2.85	1.15	1.51
(C) Contraction (1 x $\frac{1}{4}$ -in. bushing) plus 14 in. of 1-in. pipe plus 36 in. of $\frac{1}{4}$ -in. pipe				(D) Expansion (1 x $\frac{1}{4}$ -in. bushing) plus 8 in. of $\frac{1}{4}$ -in. pipe plus 61 in. of 1-in. pipe			
33.13	13.5	7.99	10.50	33.00	13.5	1.95	2.56
31.75	13.3	7.75	10.20	30.38	13.0	1.87	2.46
30.06	12.9	7.37	9.67	27.63	12.4	1.75	2.30
28.13	12.5	6.97	9.17	25.63	11.9	1.65	2.17
25.88	12.0	6.47	8.50	23.06	11.4	1.53	2.01
23.88	11.6	5.97	7.85	20.63	10.7	1.34	1.76
21.50	11.0	5.47	7.18	18.00	10.1	1.17	1.54
18.63	10.3	4.84	6.36	15.38	9.3	1.07	1.41
15.63	9.4	4.17	5.48	12.75	8.6	0.97	1.27
13.19	8.6	3.59	4.72	11.50	8.1	0.91	1.20
11.19	8.0	3.12	4.10	9.25	7.3	0.79	1.04
9.00	7.2	2.63	3.46	7.63	6.7	0.68	0.88
7.81	6.8	2.37	3.12	6.00	6.0	0.57	0.75
6.00	6.0	1.87	2.46	4.44	5.2	0.46	0.61
4.63	5.6	1.56	2.05	3.25	4.5	0.38	0.50
3.63	4.7	1.33	1.75	2.10	3.6	0.30	0.39
2.27	3.8	1.03	1.35	1.28	2.9	0.26	0.34
1.20	2.8	0.77	1.01				
0.62	2.15	0.62	0.82				

3. Measure the distances between the fittings and between the piezometer openings.

4. Determine the head loss in feet of fluid at various velocities across:
 - a. Four lengths of 1-inch pipe.
 - b. Close return bend on long straight pipe.
 - c. Contraction (1-inch to $\frac{1}{2}$ -inch).
 - d. Expansion ($\frac{1}{2}$ -inch to 1-inch).

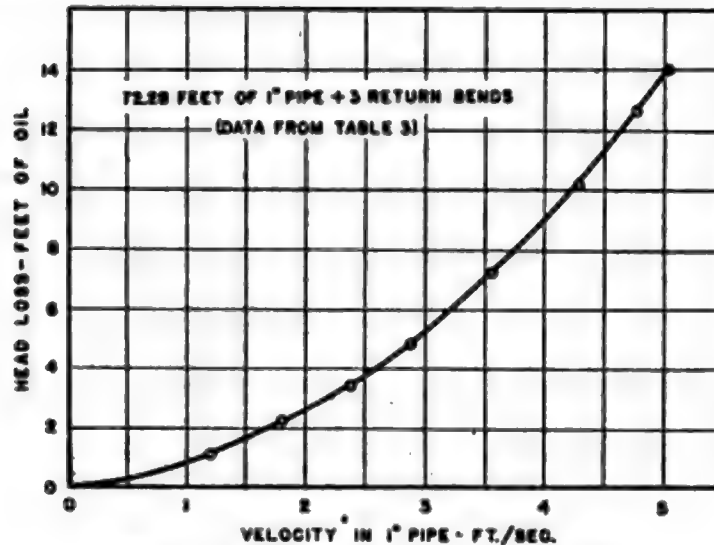


Figure 3. Head Loss in One-inch Pipe

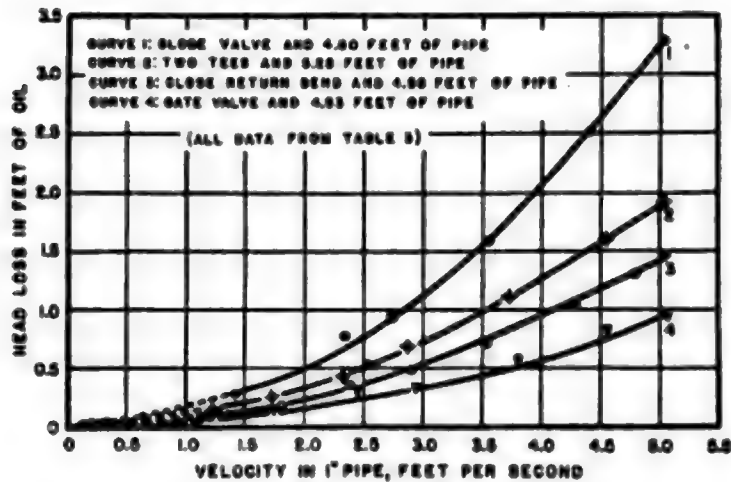


Figure 4. Head Loss in One-inch Pipe and Fittings

- e. Long section of $\frac{1}{2}$ -inch pipe.
- f. Two elbows on $\frac{1}{2}$ -inch pipe.
- g. Globe valve.
- h. Gate valve.
- i. Two elbows on 1-inch pipe.
- j. Two tees on 1-inch pipe.
- k. Seven 1-inch close return bends in series.

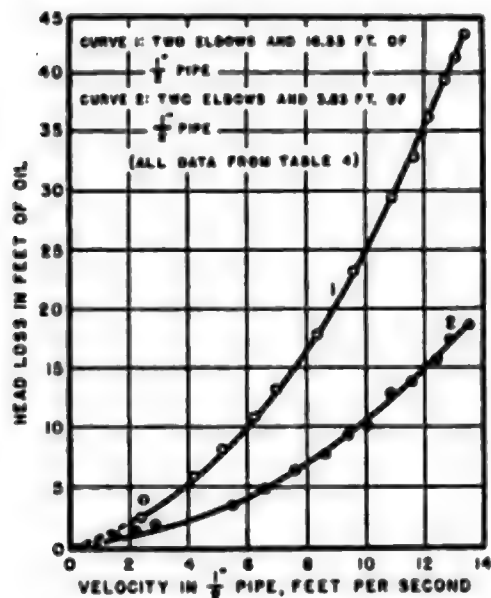


Fig. 5

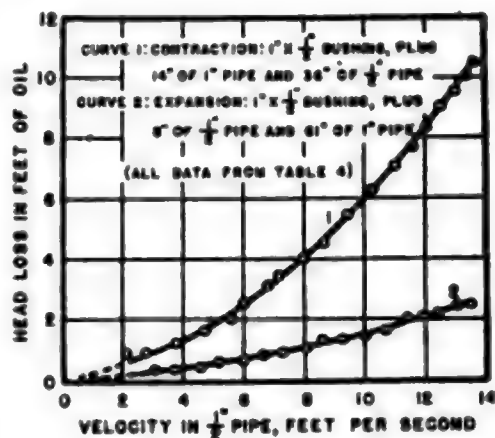


Fig. 6

Figure 5. Head Loss in One-half Inch Pipe and Elbows
Figure 6. Head Loss in Contraction and Expansion

TABLE 5

Summary of Head Losses in 1-inch Pipe and Fittings

Velocity in 1-in. pipe, ft./sec.	Head loss, ft. of oil per ft. of pipe	Equivalent length of straight pipe, ft.				
		Close return bend	Gate valve	Globe valve	Tee	Elbow
5.0	0.106	4.02	0.60	14.4	3.12	1.71
4.5	0.134	3.98	0.60	15.0	3.32	1.38
4.0	0.104	4.13	0.75	15.8	3.75	1.87
3.5	0.082	4.17	0.48	17.6	3.72	1.47
3.0	0.062	4.05	0.92	14.3	3.65	1.75
2.5	0.045	4.34	0.93	14.7	3.54	1.80
2.0	0.0296	4.25	(1.59)*	15.5	3.72	2.53
1.5	0.0181	4.32	(1.82)*	15.6	4.25	3.03
1.0	0.0089	4.48	—	17.6	5.80	—
Average.....		4.20	0.71	15.6	3.86	1.94

* Omitted in calculating average values.

5. From the experimental results, calculate:

- The head loss in feet of fluid for 1-inch pipe.
- The head loss in terms of equivalent lengths of straight pipe for all of the fittings except the contraction and expansion.
- The contraction coefficient K in the equation:

$$\Delta H_c = K \frac{u^2}{2g_0}$$

TABLE 6
Summary of Head Losses in $\frac{1}{2}$ -inch Pipe and Fittings and in Contraction and Expansion

Velocity in $\frac{1}{2}$ -in. pipe, ft./sec.	Head loss, ft. of oil per ft. of pipe	Ft. of $\frac{1}{2}$ -in. pipe equiv. to 1 elbow	Head loss for contraction, ft. of oil	$K = \frac{\Delta H_c}{u^2/2g_c}$	Head loss for expansion, ft. of oil	
					Measured	Calculated from equation $\Delta H_e = \frac{(u_2 - u_1)^2}{2g_c}$
13	1.87	2.98	1.63	0.62	2.80	1.08
12	1.70	2.54	1.22	0.55	2.23	0.94
11	1.46	2.46	1.00	0.53	2.12	0.79
10	1.19	2.55	0.98	0.63	1.84	0.65
9	0.96	2.64	0.93	0.74	1.51	0.53
8	0.80	2.46	0.72	0.72	1.20	0.42
7	0.64	2.39	0.66	0.87	1.00	0.32
6	0.48	2.51	0.64	1.15	0.82	0.23
5	0.34	2.86	0.61	1.57	0.64	0.16
4	0.22	3.20	0.60	2.42	0.48	0.10
3	0.106	(5.15)*	0.60	4.29	0.36	0.06
2	0.04	(8.10)*	0.42	6.77	0.24	0.03
Average.....		2.66				

* Omitted in calculating average values.

TABLE 7
Resistance of Screwed Fittings to the Flow of Fluids

	Equivalent length, ft. of pipe					
	1-in. close return bend	1-in. gate valve	1-in. globe valve	1-in. tee	1-in. elbow	$\frac{1}{2}$ -in. elbow
Experimental results.....	4.20	0.71	15.6	3.86	1.94	2.66
From Badger & McCabe ¹	2.34	0.57	4.68	3.11	1.57	0.84
From Crane Catalog ²	6.00	0.60	26.0	5.50	2.70	1.60

6. Compare the actual head loss for the expansion with the theoretical head loss calculated from the equation:

$$\Delta H_e = \frac{(u_2 - u_1)^2}{2g_c}$$

7. Plot the head loss in feet of fluid against the velocity in feet per second for all of the items listed under 4.

Typical Data. The experimental data and calculated results for the calibration of the Venturi meter are given in Table 2; the calibration curves are plotted in Figure 2.

A part of the data obtained in this experiment for the loss of head in pipe and fittings is given in Tables 3 and 4, and some of these data are plotted in Figures 3 to 6.

The calculated results for the head loss in the 1-inch and the $\frac{1}{2}$ -inch pipes and fittings are tabulated in Tables 5 and 6; and a comparison of some of these results with values from the literature is shown in Table 7.

SAMPLE CALCULATIONS

Item 1: *Conversion of head loss in in. of mercury to head loss in ft. of oil*

$$\begin{aligned} &= \frac{\text{in. Hg}}{12} \times \frac{(\text{sp. gr. Hg} - \text{sp. gr. oil})}{\text{sp. gr. oil}} \\ &= \frac{\text{in. Hg}}{12} \times \frac{(13.6 - 0.81)}{0.81} = \text{in. Hg} \times 1.315 \end{aligned}$$

Item 2: *Head loss per ft. of 1-in. pipe at velocity of 4.0 ft./sec.*

ΔH^* for 72.28 ft. of pipe plus 3 return bends = 8.82 (Figure 3).

ΔH for 4.58 ft. of pipe plus 1 return bend = 0.91 (Curve 3, Figure 4).

Therefore,

$$\Delta H \text{ for } 58.54 \text{ ft. of pipe} = 8.82 - 3 \times 0.91 = 6.09$$

and

$$\Delta H \text{ per ft. of 1-in. pipe} = 6.09/58.54 = 0.104$$

Item 3: *Feet of 1-in. pipe equivalent to 1 close return bend at velocity of 4.0 ft./sec.*

ΔH for 4.58 ft. of pipe plus 1 return bend = 0.91

ΔH per ft. of 1-in. pipe = 0.104 (Item 2).

Therefore,

$$\Delta H \text{ for 1 return bend} = 0.91 - 4.58 \times 0.104 = 0.48$$

and

$$\begin{aligned} \text{Feet of pipe equivalent to 1 return bend} &= 0.48/0.104 \\ &= 4.13 \end{aligned}$$

Item 4: *Feet of 1-in. pipe equivalent to 1 tee at velocity of 4.0 ft./sec.*

ΔH for 5.28 ft. of pipe plus 2 tees = 1.33 (Curve 2, Figure 4).

ΔH per ft. of pipe = 0.104 (Item 2).

* ΔH = loss of head in feet of oil.

Therefore,

$$\Delta H \text{ for 2 tees} = 1.33 - 5.28 \times 0.104 = 0.78$$

or

$$\Delta H \text{ for 1 tee} = 0.78/2 = 0.39$$

and

$$\text{Feet of pipe equivalent to 1 tee} = 0.39/0.104 = 3.75$$

Item 5: *Head loss per ft. of $\frac{1}{2}$ -in. pipe at velocity of 13 ft./sec.*

$$\Delta H \text{ for 16.33 ft. of pipe plus 2 elbows} = 41.75 \text{ (Curve 1, Figure 5).}$$

$$\Delta H \text{ for 3.83 ft. of pipe plus 2 elbows} = 18.25 \text{ (Curve 2, Figure 5).}$$

Therefore,

$$\Delta H \text{ for 12.50 ft. of pipe} = 41.75 - 18.25 = 23.50$$

and

$$\Delta H \text{ per ft. of pipe} = 23.50/12.5 = 1.87$$

Item 6: *Feet of $\frac{1}{2}$ -in. pipe equivalent to 1 elbow at velocity of 13 ft./sec.*

$$\Delta H \text{ for 3.83 ft. of pipe plus 2 elbows} = 18.25$$

$$\Delta H \text{ per ft. of pipe} = 1.87 \text{ (Item 5).}$$

Therefore,

$$\Delta H \text{ for 2 elbows} = 18.25 - 3.83 \times 1.87 = 11.10$$

or

$$\Delta H \text{ for 1 elbow} = 11.10/2 = 5.55$$

and

$$\text{Feet of pipe equivalent to 1 elbow} = 5.55/1.87 = 2.98$$

Item 7: *Contraction loss at velocity u_2 of 13 ft./sec. in $\frac{1}{2}$ -in. pipe.*

$$u_1 = \text{velocity in 1-in. pipe, ft./sec.} = 13 \times \frac{\text{area of } \frac{1}{2}\text{-in. pipe}}{\text{area of 1-in. pipe}}$$

$$= 13 \times (0.304/0.864) = 4.57$$

$$\Delta H \text{ in 1-in. to } \frac{1}{2}\text{-in. contraction plus 14 in. of 1-in. pipe plus 36 in. of } \frac{1}{2}\text{-in. pipe} = 9.7 \text{ (Curve 1, Figure 6).}$$

$$\Delta H \text{ per ft. of 1-in. pipe at 4.57 ft./sec.} = 0.138 \text{ (from plot of data in Column 2, Table 5).}$$

$$\Delta H \text{ per ft. of } \frac{1}{2}\text{-in. pipe at 13 ft./sec.} = 1.87 \text{ (Item 5).}$$

Therefore,

$$\Delta H \text{ across contraction} = 9.7 - (14/12) \times 0.138 - (36/12) \times 1.87 = 3.93$$

and

$$\Delta H_e \text{ due to contraction}$$

$$= 3.93 - \left(\frac{u_2^2 - u_1^2}{2g_c} \right) = 3.93 - \left(\frac{13^2 - 4.57^2}{64.4} \right)$$

$$= 3.93 - 2.30 = 1.63$$

Item 8: *Expansion loss at velocity of 13 ft./sec. in $\frac{1}{2}$ -in. pipe.*

ΔH in $\frac{1}{2}$ -in. to 1-in. expansion plus 8 in. of $\frac{1}{2}$ -in. pipe plus
61 in. of 1-in. pipe = 2.45 (Curve 2, Figure 6).

Therefore,

$$\Delta H \text{ across expansion} = 2.45 - (8/12) \times 1.87 - (61/12) \times 0.138 = 0.50$$

and

$$\begin{aligned} \Delta H_e \text{ due to expansion} &= 0.50 + \frac{(u_2^2 - u_1^2)}{2g_c} \\ &= 0.50 + 2.30 = 2.80 \end{aligned}$$

NOMENCLATURE

g_c = conversion factor, (lb. mass \times ft.)/(lb. force \times sec.²).

ΔH = head loss, ft. of oil.

K = contraction coefficient.

u = velocity, ft./sec.

Subscripts

1 = one-inch pipe.

2 = one-half inch pipe.

c = contraction, except in g_c .

e = expansion.

REFERENCES

1. BADGER AND McCABE, "Elements of Chemical Engineering," McGraw-Hill Book Company, New York, 2nd ed., 1936, p. 631.
2. Catalog No. 41, Crane Co., Chicago, Illinois, 1941, p. 631.

AN APPARATUS FOR DETERMINING THE PRESSURE DROP IN PIPE AND FITTINGS

Designed and Constructed by
The Department of Chemical Engineering
University of North Dakota*

Description. This fluid-flow apparatus is constructed of standard 2-inch and 1-inch galvanized pipe and screwed and welded fittings; it is arranged in five parallel sections, each approximately 20 feet long. A Venturi meter is provided in one of the sections and an orifice meter is included in the common return line. The parallel arrangement of

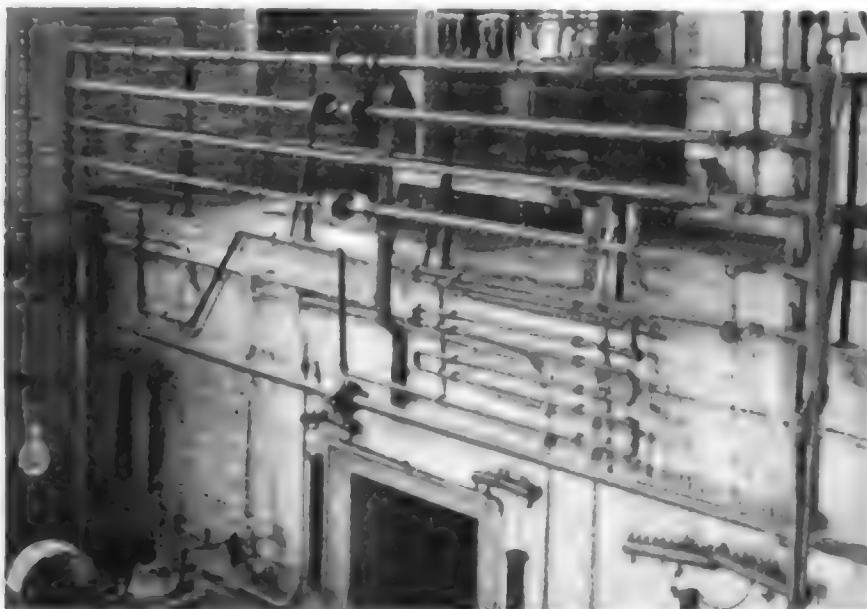


Figure 1. View of Fluid-Flow Apparatus—University of North Dakota

the various sections makes it possible to pass the fluid, by means of a centrifugal pump, through one section at a time; this permits the attainment of higher velocities than would be possible if all of the pipe and fittings were connected in series.

The pressure taps, which are provided along sections of straight pipe and before and after each fitting, are connected to common manometers by means of $\frac{3}{16}$ -inch copper tubing. For convenience in taking readings, two manometers are provided at each end of the apparatus; one manometer has mercury under water and the other has air over

* Reported by A. M. Cooley.

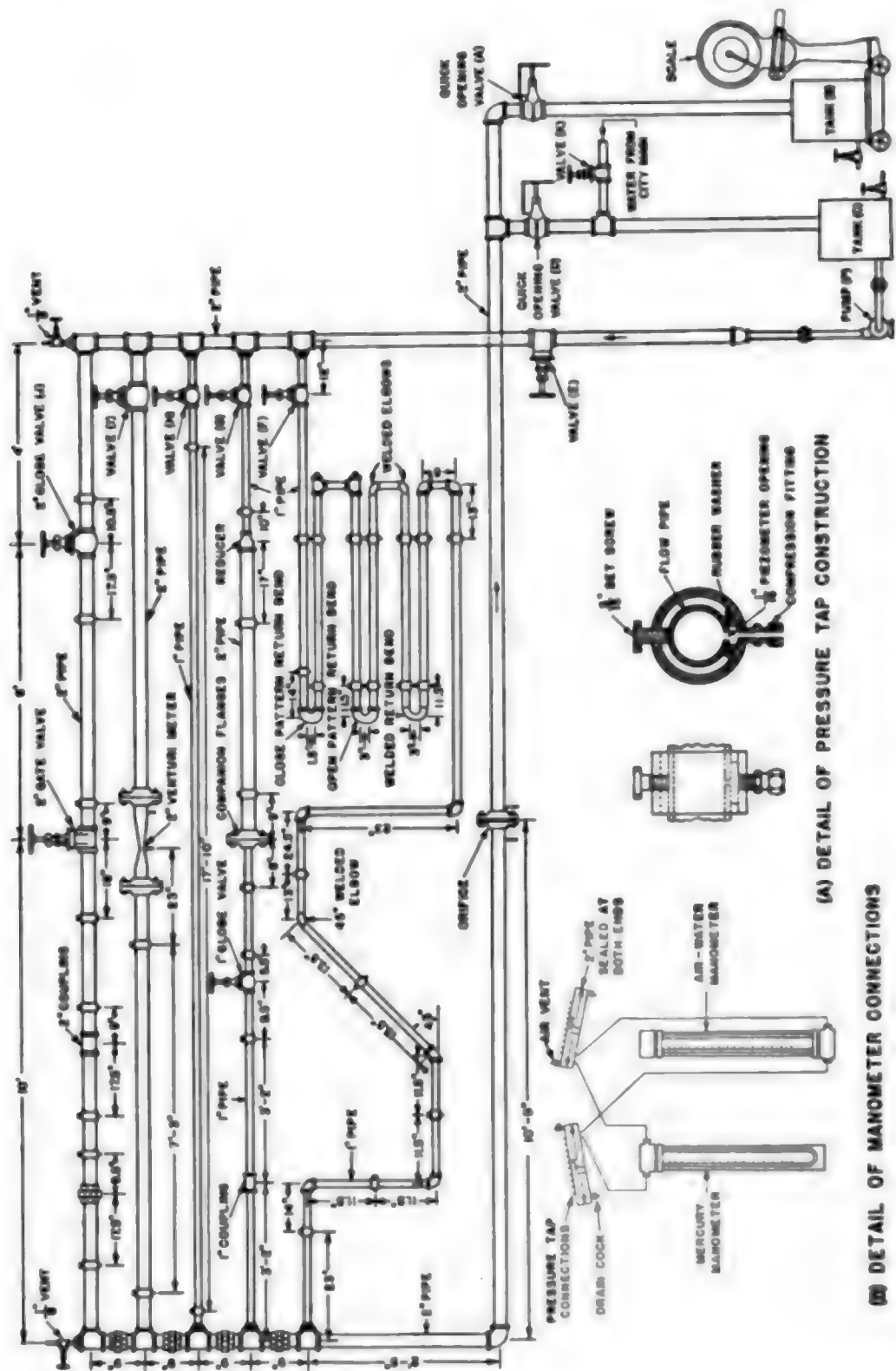


Figure 2. Apparatus for Determining Pressure Drop in Pipe and Fittings—University of North Dakota

water. The orifice and Venturi meters are not connected to the same manometers used for the pressure drop readings but are provided with individual manometers.

Figure 1 is a photograph of this fluid-flow apparatus, Figure 2 is a drawing of the equipment, and Figure 3 shows the construction of the manometers.

Materials and Cost. The cost of the materials required to construct this apparatus is approximately \$305. The necessary items are listed in Table 1.

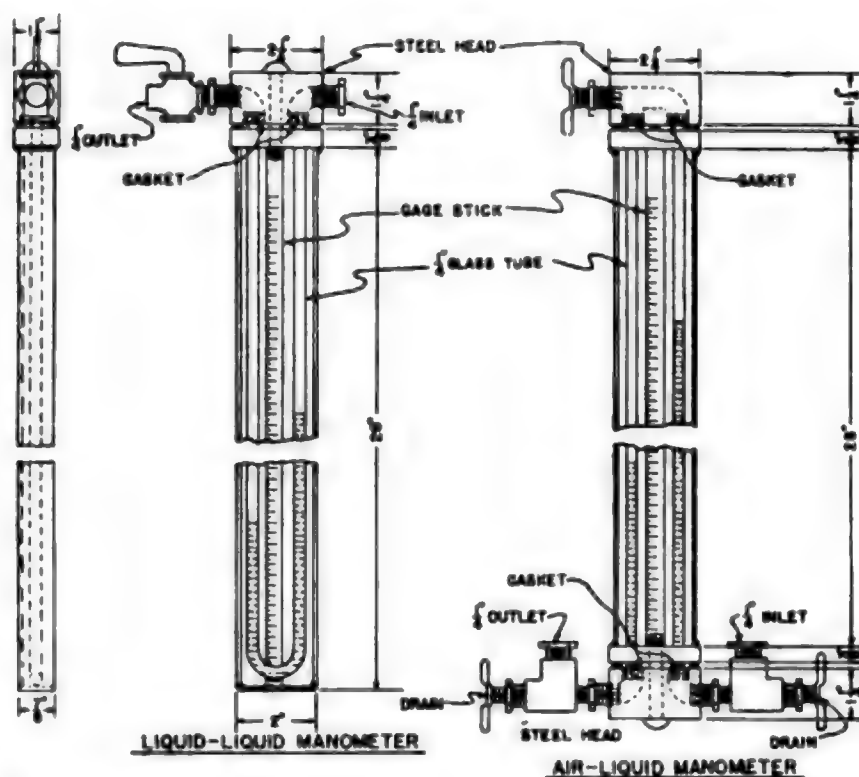


Figure 3. Details of Construction of Liquid-Liquid and Air-Liquid Manometers—University of North Dakota

Notes on Construction. 1. The pressure-tap connections were made as shown in Detail A of Figure 2. This method of construction is based on a design by the Department of Chemical Engineering, *University of Tennessee*. One-sixteenth inch holes were drilled in the pipes at the locations shown in the drawing, and the burrs were removed by means of a half-round file secured to the end of a piece of pipe. Over each opening was slipped a collar, slightly larger than the pipe and drilled and tapped to receive a $\frac{1}{16}$ -inch set screw at the top and a $\frac{1}{8}$ -inch copper to $\frac{1}{4}$ -inch male I.P.S. compression type connector at the

TABLE 1
Bill of Materials for Fluid-Flow Apparatus
 University of North Dakota

Quantity	Description	Size	Material
90 ft.	Std. pipe	2 in.	Galv. wrought steel
120 ft.	" "	1 in.	" " "
10	Std. tee	2 in.	Galv. malleable iron
1	" "	1 in.	" " "
3	Std. 90° elbow	2 in.	" " "
5	" " "	1 in.	" " "
2	" 45° "	1 in.	" " "
2	Std. butt-welding 90° elbow	1 in.	Steel
6	Ground-joint union	2 in.	Galv. malleable iron
1	Std. coupling	2 in.	" " "
1	" "	1 in.	" " "
1	Companion flange	2 x 7½ in.	Cast iron
1	" "	1 x 7½ in.	" "
1	Std. close-pattern return bend	1 in.	Galv. malleable iron
1	Std. wide-pattern return bend	1 in.	" " "
1	Std. butt-welding return bend	1 in.	Steel
1	Std. globe valve	2 in.	Brass
1	" " "	1 in.	"
4	Std. gate valve	2 in.	"
3	" " "	1 in.	"
2	" " "	½ in.	"
2	" " " (quick opening)	2 in.	"
2	Drum	55 gal.	Steel
1	Electric motor	1 H.P.	—
1	Centrifugal pump (Gould No. 1)	1 in.	All iron
1	Orifice meter	2 in. (1 in. orifice)	Cast iron, with non-corroding orifice plate
1	Venturi meter, Foxboro	2 in. (1.515 in. throat)	Cast iron
220 ft.	Tubing	⅜ in.	Copper
6	Compression tee	⅜ in.	Brass
6	Compression drain cock	⅜ in.	"
50	Compression connector	⅜ in. copper x ½ in. male I.P.S.	"
50	Straightway shut-off cock	⅜ in. copper x ½ in. male I.P.S.	"

TABLE 1—*Concluded*

Quantity	Description	Size	Material
16	Pet cock	$\frac{1}{4}$ -in. male I.P.S.	Brass
2	Manometer (air-water)	28 in. high with $\frac{1}{4}$ -in. glass tubing	Steel and glass
4	U-tube manometer (mercury)	26 in. high with $\frac{1}{4}$ in. glass tubing	" " "

bottom. A rubber gasket was inserted between the collar and the pipe at the pressure-tap opening, and the collar was then drawn up tight against the opening by means of a set screw.

The collars were made from one-inch lengths of standard pipe: those for the 2-inch lines were made from $2\frac{1}{2}$ -inch pipe and those for the 1-inch lines were made from $1\frac{1}{2}$ -inch pipe.

2. The threads of the various sections of pipe were cut with sharp dies, and the dies were lubricated with a good grade of cutting oil. All of the joints were carefully reamed to remove burrs before the pipes were assembled. No pipe dopes were used on the threaded joints; dopes are unnecessary if the threads are clean and sharp. However, a little soap solution may be used to advantage when screwing the threaded sections together.

Operation. *Calibration of Venturi and Orifice Meters.* With valves *E* and *I* open, and valves *D*, *F*, *G*, *H*, and *J* closed, water from the main is run continuously into tank *C* and pumped through the lines containing the Venturi and orifice meters; the water is discharged into tank *B* where it is weighed over timed intervals.

Several runs are made at different rates of flow by adjusting valve *E*; and calibration curves are plotted for both meters. During the pressure-drop experiments, the orifice meter is used to measure the rates of flow.

Pressure Drop in Pipe and Fittings. When measuring the pressure drop in the pipe and fittings, the liquid is first run into tank *C*; it is then circulated through the desired part of the system and returned to the tank. At each rate of flow, the pressure drop across sections of straight pipe and across various fittings are measured one by one by means of the manometers.

Typical Results. A series of runs were made with water as the test fluid. The following values were then calculated in the usual manner:

1. Orifice constant.
2. Venturi constant.

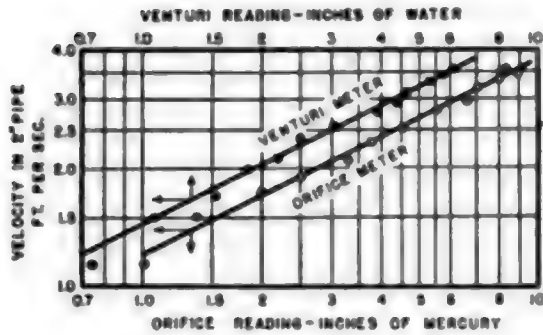


Fig. 4

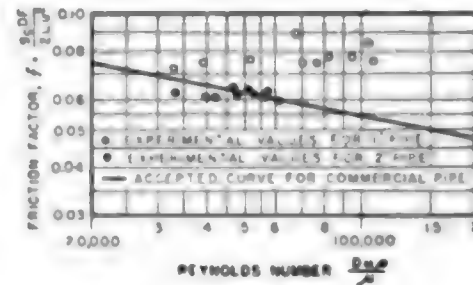


Fig. 5

Figure 4. Calibration Curves for Orifice and Venturi Meters
 Figure 5. Fanning Friction Factors for Water in Steel Pipe

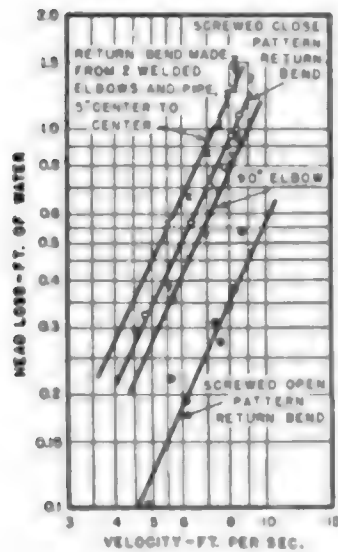


Fig. 6

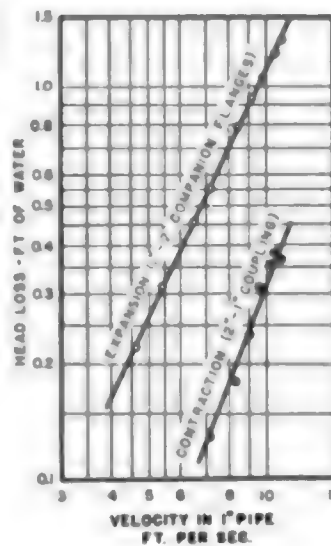


Fig. 7

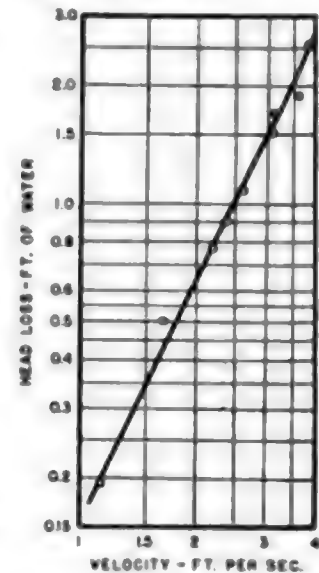


Fig. 8

Figure 6. Head Loss in One-inch Fittings
 Figure 7. Head Loss in Contraction and Expansion
 Figure 8. Head Loss in Two-inch Globe Valve

3. Head loss in feet of water per foot of straight pipe.
4. Resistance of the various fittings in terms of an equivalent length of straight pipe.
5. Head loss for contraction.
6. Head loss for expansion.
7. Contraction coefficient.
8. The friction factor in the Fanning equation.

The experimentally determined values of the head losses in pipe and fittings and in contraction and expansion, and the experimental values of the Fanning friction factor and the contraction coefficient were compared with values calculated from equations and obtained from the literature. A number of the results are shown graphically: Figure 4

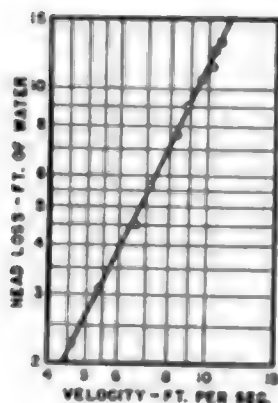


Fig. 9

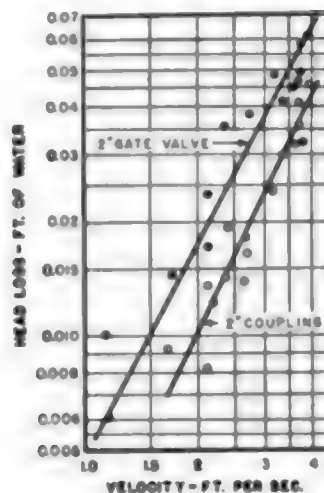


Fig. 10

Figure 9. Head Loss in One-inch Globe Valve
 Figure 10. Head Loss in Two-inch Coupling and Two-inch Gate Valve

TABLE 2

Resistance of Valves and Fittings to Flow of Fluids

Fitting	Equivalent feet of pipe		
	Experimental results	From Badger and McCabe ¹	From Crane Company ²
Std. coupling, 2-inch	1.09	—	—
Ground-joint union, 2-inch	0.25	—	—
Std. close-pattern return bend, 1-inch	2.61	2.34	0.5
Std. wide-pattern return bend, 1-inch	1.02	—	—
Std. butt-welded return bend, 1-inch	4.44	—	—
Return bend made from 2 welded elbows and pipe: 5 inches center to center	3.12	—	—
Std. butt-welded 45° elbow, 1-inch	1.97	—	—
Std. 45° elbow, 1-inch	2.19	—	1.3
Std. 90° elbow, 1-inch	2.27	1.57	2.8
Std. tee, 1-inch (through side outlet)	2.09	3.11	6
Std. globe valve, 2-inch	67.7	10.60	57
Std. gate valve, 2-inch	1.68	1.32	1.2
Std. globe valve, 1-inch	20.7	4.68	28

shows the calibration curves for the orifice and Venturi meters; Figure 5 shows the experimentally determined values of the Fanning friction factor as a function of the Reynolds' number and compares these results with the generally accepted values; and Figures 6 to 10 show the head losses for a number of fittings as a function of the velocity.

A summary of the average experimental values of the friction losses in valves and fittings in terms of the equivalent feet of pipe is given in

Table 2. For purposes of comparison, corresponding values taken from the literature are included.

REFERENCES

1. BADGER, AND McCABE, "Elements of Chemical Engineering," McGraw-Hill Book Company, New York, 2nd ed., 1936, p. 631.
2. Catalog No. 41, Crane Co., Chicago, Illinois, 1941, p. 631.

AN APPARATUS FOR FLOW OF FLUIDS THROUGH GRANULAR SOLIDS

Designed and Constructed by

The Department of Chemical Engineering
University of New Hampshire

Description. The principal parts of this apparatus are: (1) a vertical chamber, made from 1½-inch pipe, in which a bed of granular solids up to 20 inches in depth is supported on a perforated false bottom; (2) an outer steam-heated jacket made from 2-inch pipe, which is used when it is desired to heat the fluid while it is flowing through the bed; (3) pressure gages and mercury manometers for measuring the drop in pressure across the bed; and (4) an orifice meter for measuring the rate of flow of the fluid.

The apparatus is designed for use with either liquids or gases; experimentally, it has been used with water and air.

The dimensions and layout of this system are shown in Figure 1.

Materials and Cost. With the exception of the gages and the copper lines leading to the gages, the entire apparatus is constructed of standard steel pipe and fittings. These materials, which cost about \$43, are listed in Table 1.

Comments on Construction and Design. *Manometer Connections.* The use of the same manometers interchangeably for both water and air is ordinarily unsatisfactory, for: (1) after the system has been used with water it is very troublesome to free the manometer and the lines of water; and (2) after the system has been used with air it is difficult to completely displace the air with water. These difficulties have been overcome by the arrangement of traps shown in Figure 1. Regardless of whether the manometers are used with water or air, they always have water over mercury. When the system is operated with water, air in the traps and in the lines leading to the traps is permitted to escape through the cocks at the top of the traps; thus, the system is purged of air without danger of air entering the manometers or manometer lines. When the system is used with air, the upper cocks are opened and air is drawn into the traps by running water out through the lower cocks. It is necessary to have the same water level in both traps of a manometer if there is to be no zero correction. As the diameter of the traps is large compared to the diameter of the manometer tubes, any changes in the water levels during operation are insignificant.

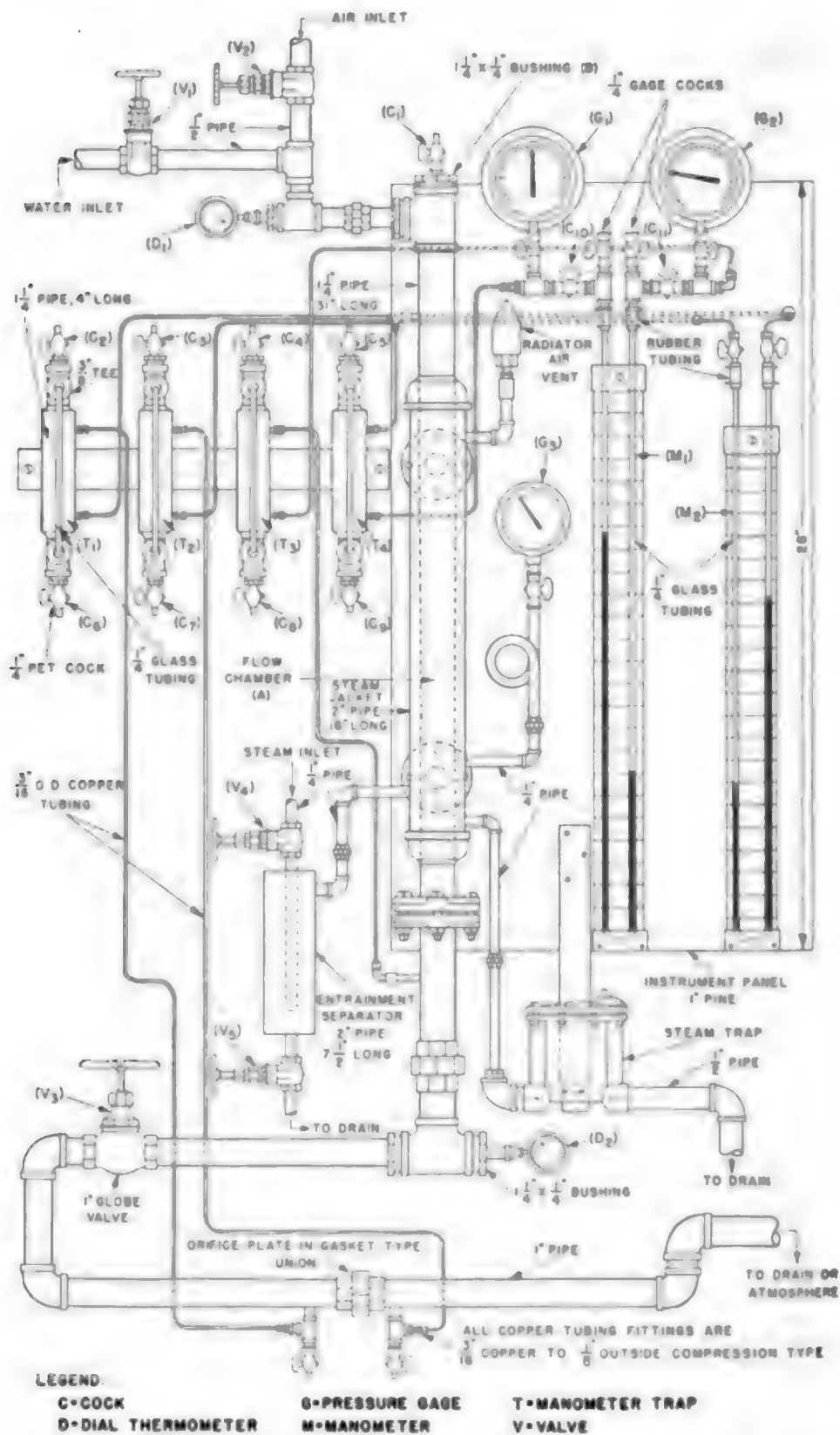


Figure 1. Apparatus for the Study of Flow of Fluids through Granular Solids—
University of New Hampshire

TABLE 1

Bill of Materials for Apparatus for Flow of Fluids through Granular Solids
University of New Hampshire

Quantity	Description	Size	Material
1½ ft.	Std. pipe	2 in.	Black wrought steel
3½ ft.	" "	1½ in.	" " "
3 ft.	" "	1 in.	" " "
6 in.	" "	1½ in.	" " "
1	Std. nipple	1½ in. x 5 in. long	" " "
1	" "	1½ in., short	" " "
1	" "	1 in. x 4 in. long	" " "
2	" "	1 in., short	" " "
1	" "	½ in. x 4½ in. long	" " "
1	" "	½ in. x 3 in. long	" " "
9	" "	½ in., short	" " "
8	" "	½ in., close	" " "
1	" "	½ in. x 6 in. long	" " "
1	" "	½ in. x 5 in. long	" " "
1	" "	½ in. x 4 in. long	" " "
2	" "	½ in. x 2 in. long	" " "
8	" "	½ in., short	" " "
15	" "	½ in., close	" " "
1	" "	½ in. x 2 in. long	" " "
1	" "	½ in., short	" " "
2	Std. tee	1½ in.	Black malleable iron
2	" "	½ in.	" " "
3	" "	½ in.	" " "
2	" "	½ in.	" " "
4	Std. 90° elbow	1 in.	" " "
2	" " "	½ in.	" " "
9	" " "	½ in.	" " "
1	Std. flange union, gasket type	1½ in.	Cast iron
1	Ground-joint union	1½ in.	Black malleable iron
1	Ground-joint union	1 in.	" " "
3	Ground-joint union	½ in.	" " "
3	Ground-joint union	½ in.	" " "
1	Bushing	1½ in. x 1 in.	Cast iron
1	"	1½ in. x ½ in.	" "
2	"	1½ in. x ½ in.	" "
2	"	½ in. x ½ in.	Malleable iron
8	"	½ in. x ½ in.	" "
5	"	½ in. x ½ in.	" "

TABLE 1—(Concluded)

Quantity	Description	Size	Material
2	Std. reducer	$\frac{1}{2}$ in. x $\frac{1}{2}$ in.	Black malleable iron
1	Std. coupling	$\frac{1}{2}$ in.	" " "
2	" "	$\frac{1}{2}$ in.	" " "
2	Std. cap	2 in.	" " "
8	" "	$\frac{1}{2}$ in.	" " "
2	Std. gate valve	$\frac{1}{2}$ in.	Brass
2	" " "	$\frac{1}{2}$ in.	"
1	Std. globe valve	1 in.	"
11	Compression-type fitting	$\frac{1}{8}$ in. copper to $\frac{1}{2}$ in. male I.P.S.	"
3	Compression-type elbow	$\frac{1}{8}$ in. copper to $\frac{1}{2}$ in. male I.P.S.	"
3	Pet cock	$\frac{1}{2}$ in. female, I.P.S.	"
8	" "	$\frac{1}{2}$ in. male, I.P.S.	"
7	Gage cock	$\frac{1}{2}$ in. female, I.P.S.	"
2	Pet cock (compression-type connection)	$\frac{1}{8}$ in.	"
2	Floor flange	$\frac{1}{2}$ in., I.P.S.	Cast iron
14 in.	Channels	$1\frac{1}{2}$ in. x $\frac{1}{2}$ in. x $\frac{1}{2}$ in.	Steel
12 in.	Flats	$1\frac{1}{2}$ in. x $\frac{1}{8}$ in.	"
12 sq. in.	Plate	$\frac{1}{2}$ in.	"
8 sq. in.	Sheet	$\frac{1}{8}$ in.	"
1	Screen, 20-mesh	$1\frac{1}{2}$ in., O.D.	Stainless steel
1	Steam gage siphon	$\frac{1}{2}$ in.	Black wrought steel
1	Steam trap, 125 lbs.	$\frac{1}{2}$ in.	Cast iron
1	Hoffman No. 4 air vent	$\frac{1}{2}$ in.	Nickel plated brass
3	Pressure gage	4 in. face; 0 to 30 lb.	Brass
10 ft.	Tubing	$\frac{1}{2}$ in.	Glass
16 ft.	"	$\frac{1}{8}$ in.	Copper
6 in.	"	$\frac{1}{2}$ in.	Rubber
5	Orifice plate	$1\frac{1}{2}$ in. O.D., x 24-gage	Brass
2	Weston dial thermometer	0 to 220°F.	Stainless steel

Orifice. The orifice was made from 1-inch pipe and a 1-inch gasket-type union. Several brass orifice plates with different sized openings are used to provide accurate readings over a considerable range of flow rates. The plates are held in the union between rubber gaskets.

Piezometer Openings. Piezometer openings in both the flow chamber and the orifice are single $\frac{1}{8}$ -inch drilled holes filed smooth on the inside of the pipes.* Steel nipples were welded over each opening, and

* Construction of piezometer openings is described on pages 27.

connections to the traps and from the traps to the gages and manometers were made with $\frac{1}{8}$ -inch copper tubing and compression-type fittings.

Support for the Solids. The perforated false bottom to hold the bed of solids is shown in detail in Figure 2. It consists of a $\frac{1}{8}$ -inch steel plate drilled with $\frac{1}{4}$ -inch holes and held between rubber gaskets in a flange union. A disk of 20-mesh stainless steel wire cloth is laid over the perforated plate to retain the bed without offering appreciable resistance to the flow.

Flow-Chamber Mounting. In order to make it easy to change the bed of solids and to clean the apparatus, the flow chamber was made

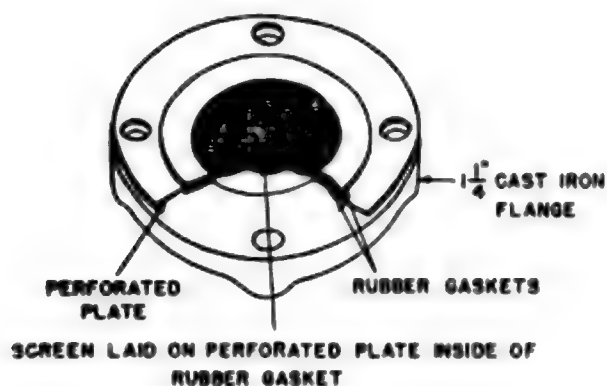


Fig. 2

Figure 2. Detail of Perforated Support for Bed of Solids

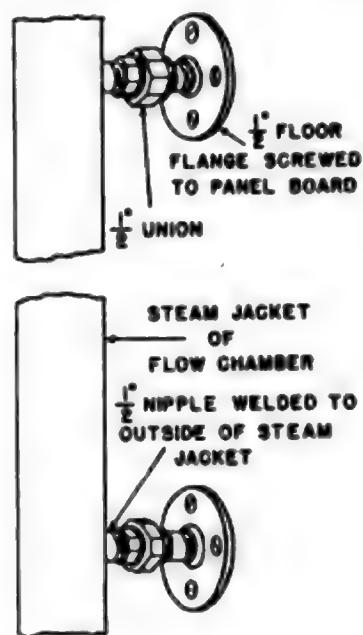


Fig. 3

Figure 3. Detail of Flow-Chamber Mounting

readily demountable. The arrangement by which it is attached to the wall is shown in Figure 3.

Operation. With Water. The chamber is filled to approximately the desired depth by removing bushing *B* and introducing the solids through the top of flow chamber *A*. The bushing is then replaced, valve V_1 is opened, and water is passed through the chamber. The pressure of the system and the rate of flow are regulated by means of valves V_1 and V_2 . If it is desired to heat the fluid, steam is introduced into the jacket through valve V_4 .

Before readings are taken, chamber *A* and traps T_1 to T_4 are purged of air through cocks C_1 to C_3 . For each run, a number of readings are

taken over the complete range of flow rates; this necessitates the changing of orifice plates.

For each rate of flow, simultaneous readings are taken of thermometers D_1 and D_2 , manometers M_1 and M_2 , and gages G_1 and G_2 . If the pressure drop becomes too great to read on manometer M_1 , cocks C_{10} and C_{11} are closed and the pressures are read on the pressure gages alone.

At the end of the run, bushing B is removed and the depth of solids in the chamber is determined by measuring the distance from the top of the tee to the top of the bed; it is necessary that this measurement be made at the end of the run because the bed usually settles appreciably after pressure is applied.

With Air. When using the system with air, the procedure is the same as that employed with water except that water is first drained from the traps until the levels are about half way in the gage glasses.

TYPICAL EXPERIMENT

Objects. 1. To determine the pressure drop of water and air through beds of sand of various particle sizes.

2. To correlate the results in a manner similar to that used for the flow of fluids through pipes.

Procedure. 1. Size a quantity of coarse sand into several fractions, using standard screens of 4 to 16 mesh.

2. Wash each fraction thoroughly with running water and then dry it.

3. Determine the percentage of voids of each fraction by displacement with water. This is done by partially filling a glass graduate with water, introducing a weighed quantity of the sand, and tapping the graduate until the sand level becomes constant. The void percentage is calculated from the level of the sand and the levels of the water before and after the introduction of the sand.

Determine the voids of each sample of sand in graduates of different diameters, both larger and smaller than the diameter of the flow chamber, and plot curves showing the effect of chamber diameter on the apparent void percentage.

4. Measure the pressure drop across beds of the various fractions with both water and air at several different velocities. In the runs with air, vary the exit pressure to study the effect of changes in the density of the air.

5. For each condition, calculate the pressure drop due to friction, and plot the results against the Reynolds' number. Base the calculations on the average screen size of the particles and the average velocity through the voids.

Experimental Results. The average particle size and the void percentage of each of the five fractions of sand used in the experiments are given in Table 2, and some of the representative experimental and calculated results for the flow of both water and air through beds of the various sand fractions are given in Tables 3 and 4. Values of the modified Fanning friction factor are plotted against the Reynolds' number in Figure 4, which includes the results of considerably more experimental data than are given in the tables.

TABLE 2
Particle Size and Void Percentage of Sand Samples

Item	Sample No.				
	1	2	3	4	5
Screen analysis:					
Through U.S.S. sieve No.....	4	6	8	10	12
Retained on U.S.S. sieve No.....	6	8	10	12	16
Average particle size, in.....	0.1595	0.1129	0.0862	0.0724	0.0565
Apparent void percentage in 1½-in. pipe (from Figure 5).....	42.6	41.7	41.3	40.9	40.6

Equations used in Calculating Friction Factors. Bernoulli's equation for the energy balance under steady flow conditions may be written as*

$$x_1 + p_1 v_1 + \frac{u_1^2}{2g_c} + \int_1^2 p dv + W = x_2 + p_2 v_2 + \frac{u_2^2}{2g_c} + F \quad (1)$$

or

$$W - \int_1^2 v dp = x_2 - x_1 + \frac{u_2^2 - u_1^2}{2g_c} + F \quad (2)$$

where

W = work input from surroundings, ft.-lb./lb.

v = specific volume, ft.³/lb.

p = absolute pressure, lb./ft.²

x = elevation above datum plane, ft.

u = velocity, ft./sec.

g_c = conversion factor = 32.174 (lb. mass × ft.)/(lb. force × sec.²).

F = loss due to friction, ft.-lb./lb.

and the subscripts 1 and 2 refer to the entrance and exit conditions, respectively.

* A more detailed treatment of these equations is given on pages 3-11.

TABLE 3
Experimental Results for Flow of Water through Beds of Sand

Depth of sand, in.	Orifice diam., in.	Temp., °F.		Pressure gage reading, lb./in. ²		Orifice manometer reading, in.	Flow rate, ft. ³ /min.	Pressure drop, in. (Hg - H ₂ O)	Viscosity, centipoises	Modified Reynolds number, <i>Re</i>	Modified Fanning friction factor, <i>f</i>
		Top	Bottom	Top	Bottom						
Fraction - 4 + 6 mesh											
13.75	0.25	52	52	—	—	13.30	0.381	18.70	1.270	1400	1.78
13.75	0.25	55	55	—	—	7.00	0.272	10.50	1.210	1050	1.96
13.75	0.25	52	52	—	—	4.20	0.210	6.85	1.270	772	2.13
13.75	0.25	60	60	—	—	1.10	0.105	2.20	1.125	446	2.75
18.88	0.25	54.5	54.5	—	—	1.10	0.105	3.00	1.220	402	2.73
18.88	0.25	55	55	—	—	0.65	0.080	2.00	1.210	310	3.13
24.44	0.25	53	53	30.0	19.0	10.20	0.334	24.20*	1.250	1259	1.68
24.44	0.25	52.5	53	—	—	8.50	0.304	21.90	1.250	1142	1.84
24.44	0.25	55	56	7.6	4.9	2.10	0.146	6.45	1.205	570	2.38
24.44	0.25	55.5	56.5	—	—	0.50	0.070	2.08	1.205	273	3.29
Fraction - 6 + 8 mesh											
21.25	0.125	52	52	12.8	9.7	13.50	0.0879	6.65	1.270	232	4.96
21.25	0.125	51	51	14.5	11.0	15.59	0.0940	7.46	1.285	246	5.07
21.25	0.125	53	53	11.0	8.1	11.39	0.0800	5.90	1.247	215	5.56
21.25	0.125	54	55	9.0	6.5	9.10	0.0722	4.98	1.220	119	5.76
21.25	0.125	50.5	50.5	16.0	12.6	17.40	0.0990	8.05	1.298	256	5.98
21.25	0.125	62	62	8.9	7.1	4.21	0.0555	3.29	1.095	171	6.35
21.25	0.125	63	63	12.8	11.6	3.25	0.0435	2.25	1.080	136	7.18
21.25	0.125	63	62.5	19.5	19.0	1.19	0.0275	1.10	1.079	86	8.79
Fraction - 8 + 10 mesh											
2.25	0.25	71	71	—	—	15.60	0.416	8.15	0.971	1111	2.02
2.25	0.25	71	71	—	—	10.10	0.330	5.45	0.971	881	2.14
2.25	0.25	72	72	—	—	6.50	0.264	3.79	0.957	716	2.33
2.25	0.25	72	72	—	—	3.70	0.195	2.30	0.957	528	2.59
2.25	0.25	72	72	—	—	0.90	0.0940	0.75	0.957	255	3.63
18.75	0.25	43	44	29.5	12.2	5.30	0.238	38.10*	1.450	423	3.46
18.75	0.25	56	58	13.2	8.5	1.05	0.102	11.16	1.171	225	5.51
18.75	0.25	44	44	4.6	1.7	0.45	0.0660	5.68	1.435	119	6.70
18.75	0.125	52	52	10.2	8.4	5.20	0.0555	4.03	1.270	113	6.72
18.75	0.125	48	48	7.6	7.0	0.30	0.0142	0.76	1.348	27.2	19.65
Fraction - 10 + 12 mesh											
17.75	0.25	44	44	30.0	8.0	6.60	0.266	48.00*	1.435	405	3.06
17.75	0.25	45	45	29.5	10.5	5.25	0.237	42.60*	1.414	369	3.42
17.75	0.25	45	46	16.7	6.1	2.60	0.164	23.87	1.400	266	4.00

TABLE 3—(Concluded)

Depth of sand, in.	Orifice diam., in.	Temp., °F.		Pressure gage reading, lb./in. ²		Orifice manometer reading, in.	Flow rate, ft. ³ /min.	Pressure drop, in. (Hg - H ₂ O)	Viscosity, centipoises	Modified Reynolds number, Re	Modified Fanning friction factor, f
		Top	Bottom	Top	Bottom						
Fraction - 10 + 12 mesh—(Concluded)											
17.75	0.25	50	50	6.7	4.1	0.35	0.0580	5.10	1.308	98	6.85
17.75	0.125	46	46	17.7	12.6	17.61	0.100	11.96	1.391	158	5.40
17.75	0.125	48	50	18.7	15.1	9.20	0.0730	7.66	1.328	121	6.50
17.75	0.125	57	58	6.5	4.5	3.31	0.0450	4.74	1.165	85	10.57
17.75	0.125	46	46	24.8	24.1	0.42	0.0169	1.00	1.391	26.8	15.80
Fraction - 12 + 16 mesh											
14.00	0.125	58	58	18.3	13.0	17.60	0.100	12.88	1.159	150	5.61
14.00	0.125	76.5	76	13.0	12.0	1.20	0.0275	1.98	0.903	52.2	11.40
14.00	0.125	62	64	15.9	15.1	0.26	0.0132	0.85	1.085	21.0	21.40
14.00	0.25	56	56.5	26.0	6.7	5.10	0.231	42.50*	1.187	337	3.47
24.38	0.25	43	42.5	21.3	13.0	10.00	0.0760	19.40	1.475	88.9	8.40
24.38	0.25	43	43	16.9	10.1	7.05	0.0640	15.10	1.460	75.3	9.22
24.38	0.25	52	53	25.0	20.8	3.90	0.0485	9.82	1.260	66.4	10.43
24.38	0.25	56	56	20.0	17.5	1.58	0.0311	5.85	1.191	40.4	15.12
24.38	0.25	45	44	8.8	7.1	0.60	0.0200	3.25	1.427	24.2	20.30
24.38	0.25	45.5	45	19.7	18.3	0.37	0.157	2.50	1.408	19.2	25.39

* Values calculated from pressure gage readings.

The friction loss F includes not only the loss in straight pipe but also any losses due to valves and fittings and contraction and expansion.

Under the conditions of the experiment: (1) no work was done on the fluid and, therefore, $W = 0$; (2) the gages were at the same elevation, and, therefore, $x_1 = x_2$; (3) with water flowing through the system, $u_1 = u_2$ and with air flowing, $(u_2^2 - u_1^2)$ was negligible; and (4) although there were some losses due to contraction and expansion when the fluid entered and left the bed of sand, these losses were negligible compared to the loss due to friction through the bed itself.

For this experiment, therefore, Equations (1) and (2) become:

$$F = p_1 v_1 - p_2 v_2 + \int_1^2 p \, dv \quad (3)$$

or

$$F = - \int_1^2 v \, dp = \int_2^1 v \, dp \quad (4)$$

where F represents only the loss due to friction through the bed of sand.

TABLE 4
Experimental Results for Flow of Air Through Beds of Sand

Depth of sand, in.	Orifice diam., in.	Temp., °F.		Pressure gage reading, lb./in. ²		Orifice manometer reading, in.	Flow rate, ft. ³ /min. at 75°F. and 14.5 lb./in. ² abs.	Pressure drop, in. (Hg - H ₂ O)	Modified Reynolds' number, <i>R_s</i>	Modified Fanning friction factor, <i>f</i>
		Top	Bottom	Top	Bottom					
-4 + 6 mesh										
12.44	0.25	75	78	9.0	7.4	3.0	8.75	2.95	2610	0.786
12.44	0.25	75.5	77	7.5	4.7	5.0	11.60	5.70	3458	0.786
12.44	0.25	75.5	76.5	8.0	5.5	5.8	12.50	6.55	3725	0.798
12.44	0.125	75	76.5	17.9	17.6	1.0	1.28	0.10	381	1.762
12.44	0.125	75	76	18.6	18.3	3.0	2.19	0.20	552	1.231
12.44	0.125	75	76	17.4	16.9	5.0	2.81	0.36	837	1.294
12.44	0.125	75	76	14.4	13.8	9.0	3.76	0.70	1120	1.270
12.44	0.125	75	76	10.6	9.8	15.1	4.84	1.40	1443	1.320
-6 + 8 mesh										
13.75	0.25	75	77	9.6	6.9	3.0	8.75	5.70	1985	0.928
13.75	0.25	75	77	9.7	5.2	5.0	11.60	9.75	2500	0.873
13.75	0.25	75	76	11.7	6.0	7.0	13.80	12.80	2970	0.862
13.75	0.125	74.5	77	18.1	18.0	1.0	1.28	0.20	276	2.180
13.75	0.125	75	77	18.9	18.3	5.0	2.81	0.67	606	1.541
13.75	0.125	74.5	78	16.4	15.6	9.0	3.76	1.20	801	1.420
13.75	0.125	74	78	12.9	11.6	13.0	4.50	2.00	970	1.450
13.75	0.125	74	78	11.4	10.0	17.0	5.11	2.70	1102	1.429
-8 + 10 mesh										
11.75	0.25	73	71.5	15.5	14.6	0.4	3.00	1.00	498	1.581
11.75	0.25	71	71	11.0	8.0	3.0	8.75	6.30	1454	0.951
11.75	0.25	72.5	70.5	10.0	4.9	5.0	11.60	11.10	1925	0.872
11.75	0.25	73	70.5	11.7	5.5	6.4	13.10	13.95	2171	0.904
11.75	0.125	73	73	11.7	11.1	3.2	2.27	0.70	377	1.695
11.75	0.125	73.5	72.5	14.9	14.0	7.0	3.32	1.20	551	1.517
11.75	0.125	74	73	11.9	10.6	11.4	4.21	2.20	700	1.540
11.75	0.125	74	73	14.0	12.5	17.0	5.11	3.00	850	1.532
-10 + 12 mesh										
13.50	0.25	75	76	11.9	10.8	0.55	3.55	1.80	500	1.275
13.50	0.25	75.5	76	14.4	8.5	5.0	11.60	12.45	1635	0.828
13.50	0.125	76	77	16.6	16.4	1.0	1.28	0.35	181	2.280
13.50	0.125	75	77	12.2	11.7	3.0	2.19	0.80	310	1.522
13.50	0.125	75.5	77	15.2	14.5	5.0	2.81	1.15	396	1.475
13.50	0.125	75	77	16.7	15.5	9.1	3.79	1.90	534	1.396
13.50	0.125	75.5	76.5	17.6	16.1	13.0	4.50	2.60	634	1.386
13.50	0.125	76	76	16.6	14.6	17.0	5.11	3.45	721	1.370

TABLE 4—Concluded

Depth of sand, in.	Orifice diam., in.	Temp., °F.		Pressure gage reading, lb./in. ²		Orifice manometer reading, in.	Flow rate, ft. ³ /min. at 75°F. and 14.5 lb./in. ² abs.	Pressure drop, in. (Hg — H ₂ O)	Modified Reynolds' number, <i>Re</i>	Modified Fanning friction factor, <i>f</i>
		Top	Bottom	Top	Bottom					
-12 + 16 mesh										
14.06	0.25	74	71	11.5	5.4	3.1	8.90	13.40	987	0.987
14.06	0.25	73	73.5	16.7	10.7	3.8	9.95	13.20	1105	0.956
14.06	0.25	74	72	13.5	11.3	5.0	11.60	20.40	1286	1.040
14.06	0.125	75	74	18.9	17.5	1.0	1.28	0.40	142	2.030
14.06	0.125	75	74	16.5	15.5	3.0	2.19	1.05	243	1.697
14.06	0.125	75	74	16.1	15.3	5.0	2.81	1.50	312	1.460
14.06	0.125	75	74	17.7	16.5	9.0	3.76	2.45	416	1.395
14.06	0.125	74.5	74	16.7	14.5	15.0	4.82	4.50	534	1.448

* Calculations were based on an average temperature of 75°F. for all of the runs. The viscosity at this temperature is 0.0183 centipoises.

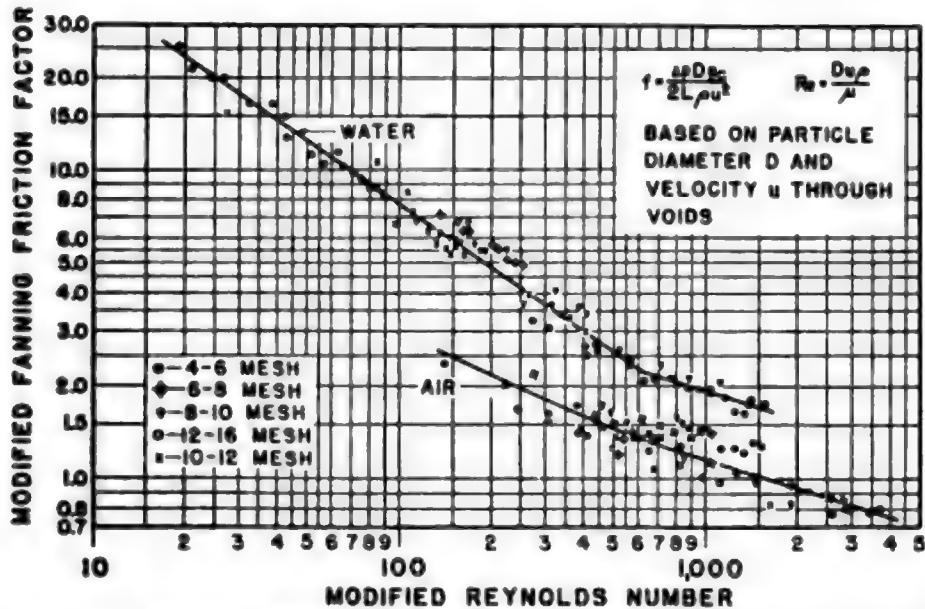


Figure 4. Modified Friction Factor for Flow of Water and Air through Beds of Sand

For liquids, $v_1 = v_2$ and $\int p dv = 0$; therefore, Equations (3) and (4) reduce to the simple expression

$$F = v\Delta p = \frac{\Delta p}{\rho} \quad (5)$$

where

Δp = the pressure drop due to friction, lb./ft.²

ρ = density, lb./ft.³

For gases, v changes with the pressure, but if the flow is substantially isothermal and the pressure is sufficiently low* so that ideal gas laws may be assumed, Equation (4) may be written as

$$F = 2.303RT \log_{10} \frac{p_1}{p_2} \quad (6)$$

where

R = gas constant, ft.-lb./(lb. \times °R.).

T = absolute temperature, °R.

Furthermore, if the pressure drop is sufficiently low, Equation (4) may be simplified to

$$F = v_{av} \Delta p = \frac{\Delta p}{\rho_{av}} \quad (7)$$

where v_{av} and ρ_{av} are the arithmetic averages of v_1 and v_2 , and ρ_1 and ρ_2 , respectively.

Friction loss data are usually correlated by means of the Fanning equation

$$F = \frac{4fLu^3}{2g_c D} \quad (8)$$

where

f = Fanning friction factor, dimensionless.

L = length of the duct, ft.

u = velocity, ft./sec.

g_c = conversion factor = 32.174 (lb. mass \times ft.)/(lb. force \times sec.²).

D = diameter, ft.

Substituting this value of F in Equations (5), (6), and (7) and solving for f :

1. For liquids

$$f = \frac{v\Delta p D g_c}{2Lu^3} = \frac{\Delta p D g_c}{2L\rho u^3} \quad (9)$$

2. For gases at low pressure drop

$$f = \frac{2.303RTg_c D}{2Lu_{av}^3} \log_{10} \frac{p_1}{p_2} \quad (10)$$

* The limitations of the flow equations are discussed on pages 9-11.

or

$$f = \frac{v_{av} \Delta p D g_c}{2 L u_{av}^2} = \frac{\Delta p D g_c}{2 L \rho_{av} u_{av}^2} \quad (11)$$

Where the pressure drop for the flow of gases is greater than about 10 per cent of the initial pressure, it is advisable to take into consideration the work of expansion in calculating the friction factor from pressure drop measurements. For the isothermal flow of gases where velocity effects are negligible, this equation is*

$$f = \frac{g_c A^2 D}{4 L w^2 R T} (p_1^2 - p_2^2) \quad (12)$$

where

A = cross-sectional area of the fluid stream, ft.²

w = flow rate, lb./sec.

and f , D , g_c , p , L , R and T have the same units as in Equations (1) to (11).

The value of the friction factor in all of the above equations varies with the dimensionless modified Reynolds' number

$$Re = \frac{D u \rho}{\mu} \quad (13)$$

where

D = average size of the sand particles.

u = velocity through the voids.

ρ = density of the fluid.

μ = viscosity of the fluid.

When using any of the equations for flow through porous media, difficulties are encountered because the values of D and u are not known. In the treatment presented here, D is taken as the mean screen size of the sand particles, and u is taken as the velocity through the voids; the latter value is equal to the fraction of voids times the velocity based on the cross-sectional area of the empty flow chamber. This treatment is somewhat different from that of Chilton and Colburn¹ who correlated numerous data on the basis of the velocity through the empty chamber and included a wall effect factor computed by Furnas.²

Determination of Void Percentage. In determining the percentage of voids in a mass of granular solids, it is necessary to take into consideration the size of the container in which the determination is made, for the conditions at the contact surface are different than they are in the interior of the mass. As the particle diameter decreases and the con-

* The derivation of this equation is given on page 10.

tainer diameter increases, the wall effect becomes less pronounced and the apparent void percentage decreases, as is shown by the curves in Figure 5 which were plotted from experimental data. The various curves tend to approach a common value, which is the true void percentage and, therefore, they should be concave upward and asymptotic to a horizontal line representing the true void percentage. However, over the range involved in this experiment, the data were substantially represented by straight lines.

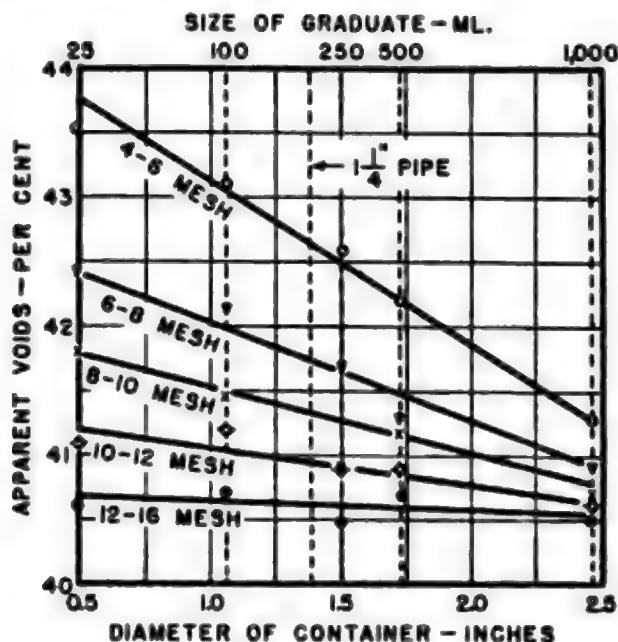


Figure 5. Apparent Void Percentage as a Function of Particle Size and Container Diameter

The calculations involved in the determination of each point on these curves are illustrated by the following sample calculations:

DATA:

Sand fraction	= -4 + 6 mesh
Mean particle size	= 0.1595 in.
Weight of sand	= 126.2 grams
Volume of water	= 50.0 ml.
Volume of water + sand	= 98 ml.
Volume of sand + voids	= 84.5 ml.

CALCULATIONS:

True volume of sand	= 98 - 50 = 48 ml.
Volume of the voids	= 84.5 - 48 = 36.5 ml.
Apparent void percentage	= $36.5/84.5 \times 100 = 43.2$
True density of the sand	= $126.2/48 = 2.63$ grams/ml.
Bulk density of the sand	= $126.2/84.5 = 1.49$ grams/ml.

Calculation of the Friction Factor and the Reynolds' Number for the Flow of Water. These calculations were made by means of Equations (9) and (13) using the data in Table 3. The method of computation is illustrated by the calculations for the first set of readings with the -4 + 6 mesh sand (first line of Table 3).

DATA:

Sand Fraction	= -4 + 6 mesh
Depth of bed	= 13.75 in. = 1.146 ft. = L
Percentage of voids	= 42.6
Average size of particles	= 0.1595 in. = 0.01330 ft. = D
Orifice reading	= 13.30 in. (water over mercury)
Flow rate	= 0.381 cu.ft./min. = V (from orifice calibration)
Pressure drop	= 18.70 in. (water over mercury) = r
Viscosity	= 1.270 centipoises at 52° F. = $1.270 \times 0.000672 = 0.000853$ lb./(ft. \times sec.) = μ
Cross-sectional area of flow chamber	= 1.495 in. ² = 0.01038 ft. ² = a
Specific gravity of mercury	= 13.6
Specific gravity of water	= 1.0
Density of water	= 62.4 lb./ft. ³ = ρ
Conversion factor	= 32.174 (lb. mass \times ft.)/(lb. force \times sec. ²) = g_c

CALCULATIONS:

1. *Fanning friction factor.*

$$\begin{aligned} \text{a. } \Delta p &= r \times \frac{(\text{sp. gr. of Hg} - \text{sp. gr. H}_2\text{O})}{12} \times \rho \\ &= 18.70 \times \frac{(13.6 - 1.0)}{12} \times 62.4 = 1230 \text{ lb./ft.}^2 \end{aligned}$$

$$\begin{aligned} \text{b. } u &= \frac{V}{60} \times \frac{1}{a \times \frac{\text{percentage of voids}}{100}} \\ &= \frac{0.381}{60} \times \frac{1}{0.01038 \times \frac{42.6}{100}} = 1.435 \text{ ft./sec.} \end{aligned}$$

Therefore,

$$f = \frac{\Delta p D g_c}{2 L \rho u^2} = \frac{(12.30)(0.01330)(32.174)}{(2)(1.146)(62.4)(1.435)^2} = 1.78$$

2. *Reynolds' number.*

$$Re = \frac{D u \rho}{\mu} = \frac{(0.01330)(1.435)(62.4)}{0.000853} = 1400$$

Calculation of the Friction Factor and Reynolds' Number for Flow of Air. In order to compare the various equations for the flow of gases, the friction factor for air was calculated by means of Equations (10), (11) and (12). In all cases, even with the maximum pressure drop encountered in this experiment, the results of the three sets of calculations did not differ by more than a few tenths of one per cent, which was well within slide rule accuracy. The calculations for the first set of readings (first line of Table 4) are as follows:

DATA:

Sand fraction	= -4 + 6 mesh
Depth of bed	= 12.44 in. = 1.037 ft. = L
Percentage of voids	= 42.6
Average size of particles	= 0.1595 in. = 0.01330 ft. = D
Temp. of entering air	= 75°F.
Temp. of exit air	= 78°F.
Pressure entering air	= 9.0 lb./in. ² gage = P_1
Pressure exit air	= 7.4 lb./in. ² gage = P_2
Orifice reading	= 3.0 in. (water over mercury)
Flow rate	= 8.75 ft. ³ /min. at 75°F. and 14.5 lb./in. ² = V (from orifice calibration)
Pressure drop	= 2.95 in. (water over mercury) = r
Viscosity	= 0.0183 centipoises at 75°F.* = $0.0183 \times 0.000672 = 0.00001230$ lb./(ft. \times sec.) = μ
Barometric pressure	= 750 mm. Hg = 14.5 lb./in. ² = 2088 lb./ft. ² = p_b
Cross-sectional area of flow chamber	= 1.495 in. ² = 0.01038 ft. ² = a
Specific gravity of mercury	= 13.6

* All of the calculations for air were based on a temperature of 75°F. which was the average temperature for all of the runs. The maximum deviation from this value was only a few degrees and this would have a negligible effect on the results.

Specific gravity of water	= 1.0
Density of water	= 62.4 lb./ft. ³ = ρ_w
Conversion factor	= 32.174 (lb. mass \times ft.)/(lb. force \times sec. ²) = g_c
Gas constant	= 53.3 ft.-lb./lb. \times °R. = R
Absolute temperature	= 75 + 460 = 535°R.

CALCULATIONS:

1. *Average pressure of the air*

$$= \frac{1}{2}(P_1 + P_2) + 14.5 = \frac{1}{2}(9.0 + 7.4) + 14.5 = 22.7 \text{ lb./in.}^2 \text{ abs.}$$

$$= 22.7 \times 144 = 3269 \text{ lb./ft.}^2 \text{ abs.} = p_{av}$$

2. *Pressure drop*

$$= \frac{r(\text{sp. gr. Hg} - \text{sp. gr. H}_2\text{O})}{12} \times \rho_w = \frac{2.95(13.6 - 1.0)}{12} \times 62.4$$

$$= 193.3 \text{ lb./ft.}^2 = \Delta p$$

3. *Entering and exit pressures calculated from the average pressure p_{av} and the pressure drop Δp .*

$$\text{a. Entering pressure} = p_{av} + \frac{1}{2}\Delta p = 3269 + \frac{1}{2}(193.3)$$

$$= 3365.5 \text{ lb./ft.}^2 \text{ absolute} = p_1$$

$$\text{b. Exit pressure} = p_{av} - \frac{1}{2}\Delta p = 3269 - \frac{1}{2}(193.3)$$

$$= 3172.5 \text{ lb./ft.}^2 \text{ absolute} = p_2$$

4. *Average specific volume of the air*

$$= \frac{359}{29} \times \frac{535}{492} \times \frac{14.7}{22.7} = 8.71 \text{ ft.}^3/\text{lb.} = v_{av}$$

where

359 = ft.³/lb. mol at 492°R. and 14.7 lb./in.² absolute.

29 = average molecular weight of air, lb./mol.

535 = absolute temp. of the air, °R.

22.7 = average pressure of the air, lb./in.² absolute.5. *Effective cross-sectional area of fluid stream*

$$= a \times \frac{\text{percentage of voids}}{100} = 0.01038 \times \frac{42.6}{100} = 0.00442 \text{ ft.}^2 = A$$

6. *Average velocity of the air*

$$\begin{aligned}
 &= \frac{V}{60} \times \frac{1}{A} \times \frac{p_b}{p_{av}} \\
 &= \frac{8.75}{60} \times \frac{1}{0.00442} \times \frac{2088}{3269} = 21.0 \text{ ft./sec.} = u_{av}
 \end{aligned}$$

7. *Mass flow rate*

$$= \frac{u_{av}}{v_{av}} \times A = \frac{21.0}{8.71} \times 0.00442 = 0.01066 \text{ lb./sec.} = w$$

8. *Fanning friction factor by Equation (10).*

$$\begin{aligned}
 f &= \frac{2.303RTg_c D}{2Lu_{av}^2} \times \log_{10} \frac{p_1}{p_2} \\
 &= \frac{2.303 \times 53.3 \times 535 \times 32.174 \times 0.01330}{2 \times 1.037 \times (21.0)^2} \times \log_{10} \frac{3365.5}{3172.5} \\
 &= 0.788
 \end{aligned}$$

9. *Fanning friction factor by Equation (11).*

$$\begin{aligned}
 f &= \frac{v_{av} \Delta p D g_c}{2Lu_{av}^2} \\
 &= \frac{8.71 \times 193.3 \times 0.01330 \times 32.174}{2 \times 1.037 \times (21.0)^2} \\
 &= 0.786
 \end{aligned}$$

10. *Fanning friction factor by Equation (12).*

$$\begin{aligned}
 f &= \frac{g_c A^2 D}{4Lw^2 RT} (p_1^2 - p_2^2) \\
 &= \frac{32.174 \times (0.00442)^2 \times 0.01330}{4 \times 1.037 \times (0.01066)^2 \times 53.3 \times 535} [(3365.5)^2 - (3172.5)^2] \\
 &= 0.783
 \end{aligned}$$

11. *Reynolds' number.*

$$Re = \frac{Du\rho}{\mu} = \frac{Du}{\mu v_{av}} = \frac{0.01330 \times 21.0}{0.00001230 \times 8.71} = 2610$$

Comments on Results. When the experimentally determined values of the Fanning friction factor for the flow of water were plotted against the modified Reynolds' number on log-log paper (Figure 4), the points for all of the sand fractions fell reasonably close to two straight lines

which intersect at a Reynolds' number of approximately 600, indicating that the flow changes from viscous to turbulent at about this point.

The results for air, however, were not as satisfactory; the scattering of points was greater, the critical Reynolds' number could not be determined, and the curve was considerably below the curve for water.

It is probable that the poorer results with air were due to differences in the packing of the sand particles. When the sand was lubricated with water, the frictional resistance between particles was lowered and there was a greater tendency for them to pack together under pressure and thereby occupy a volume which would have a minimum of voids. With air flowing through the bed, there was less tendency toward packing, and, as a result, the percentage of voids was probably appreciably greater than that determined experimentally. This would result in actual velocities through the voids much lower than those calculated, and this, in turn, would be reflected in calculated values of f much lower than the actual values. This is evident from Equations (11) and (13), for in the friction factor equation the velocity enters in the denominator and is squared, while in the Reynolds' number, although the velocity is in the numerator, it enters only as the first power.

NOMENCLATURE

- A = effective cross-sectional area of fluid stream, ft.²
- a = cross-sectional area of empty flow chamber, ft.²
- D = average size of sand particles, ft.
- f = Fanning friction factor, dimensionless.
- F = loss due to friction, ft.-lb./lb.
- g_c = conversion factor = 32.174 (lb. mass \times ft.)/(lb. force \times sec.²).
- L = length of duct, or depth of bed, ft.
- P = pressure, lb./in.² gage.
- p = absolute pressure, lb./ft.²
- Δp = pressure drop due to friction, lb./ft.²
- r = pressure drop, in. (water over mercury).
- R = gas constant, ft.-lb./(lb. \times °R.).
- Re = modified Reynolds' number, dimensionless.
- T = absolute temperature, °R.
- u = velocity through voids, ft./sec.
- v = specific volume, ft.³/lb.
- V = flow rate, ft.³/min.
- w = mass flow rate, lb./sec.
- W = work input from surroundings, ft.-lb./lb.
- x = height above datum plane, ft.

μ = viscosity, lb./ (ft. \times sec.).

ρ = density, lb./ft.³

Subscripts

1 = entrance condition.

2 = exit condition.

av = average.

b = atmospheric condition.

w = water.

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SECTION 2

FLOW OF HEAT

INTRODUCTION

A study of the flow of heat is one of the most important phases of chemical engineering, for practically every operation or process involves the transfer of heat, either directly or indirectly. The importance of heat flow is evident in evaporation, drying, and distillation, but heat may also be of importance in such operations as crushing and grinding, and fluid flow.

Heat may be transferred by conduction, by radiation, and by convection, but in most cases of practical importance flow takes place as the result of two or more of these basic mechanisms operating simultaneously. For example, in an insulated pipe carrying hot water, heat is transferred to the inner film by convection and then passes through the inner film, the metal wall, and the insulation by conduction; part of this heat is then radiated from the surface of the insulation while the rest passes through the air film by conduction. Beyond the air film, convection again becomes important.

HEAT TRANSFER BY CONDUCTION

The equation for the transfer of heat by conduction may be written as

$$\frac{Q}{\theta} = \frac{\Delta t}{R} \quad (2.1)$$

where

Q = heat transferred in time θ .

Δt = temperature difference.

R = thermal resistance.

If the heat is being transferred through a homogeneous medium, the resistance is defined by *Fourier's law* as

$$R = \frac{L}{kA} \quad (2.2)$$

where

L = distance across which Δt is measured.

A = area at right angles to the flow of heat.

k = thermal conductivity of the medium.

If the heat is flowing through more than one medium in series, the

total resistance is the sum of all the resistances, and Equation (2.1) becomes

$$\frac{Q}{\theta} = \frac{\Delta t}{R_1 + R_2 + R_3 + R_4 + \dots} \quad (2.3)$$

$$= \frac{\Delta t}{\frac{L_1}{k_1 A_1} + \frac{L_2}{k_2 A_2} + \frac{L_3}{k_3 A_3} + \frac{L_4}{k_4 A_4} + \dots}$$

where Δt is the overall temperature drop.

In the usual heat exchanger, heat is transferred from one fluid to another through a metal wall, and only the temperatures of the two fluids, but not the temperatures of the wall surfaces, are known. In this case, three resistances are involved: the resistance of the wall, and the resistances of the two fluid films on the surfaces of the wall. However, as the thicknesses of the films are seldom known, the quantities L and k are combined and a new term is defined by the equation

$$h = \frac{k}{L} \quad (2.4)$$

where h is the film coefficient of heat transfer.

Equation (2.3) then becomes

$$\frac{Q}{\theta} = \frac{\Delta t}{\frac{1}{h_1 A_1} + \frac{L}{k A_m} + \frac{1}{h_2 A_2}} \quad (2.5)$$

where, in the usual units

Q/θ = heat transferred, B.t.u./hr.

Δt = difference in temperature between one fluid and the other, °F.

k = thermal conductivity of the metal wall, B.t.u.-ft./
(hr. \times ft.² \times °F.).

h_1 and h_2 = inner- and outer-film coefficients, respectively,
B.t.u./(hr. \times ft.² \times °F.).

A_1 , A_m and A_2 = inner area, mean area, and outer area of the
metal wall, respectively, ft.²

If the heat is flowing through the wall of a pipe of appreciable thickness compared to its diameter, A_m is calculated by means of the logarithmic-mean equation

$$A_m = \frac{A_2 - A_1}{2.303 \log_{10} \frac{A_2}{A_1}} = \frac{D_2 - D_1}{2.303 \log_{10} \frac{D_2}{D_1}} \quad (2.6)$$

where D_1 and D_2 are the inner and outer diameters of the pipe, respectively, ft.

In most cases, however, it is unnecessary to use the log-mean area; for if the ratio D_2/D_1 is less than about 2 the arithmetic mean is usually sufficiently accurate.

Overall Coefficients. If both the numerator and denominator of the right hand side of Equation (2.5) are multiplied by either A_1 , A_m , or A_2 , the equation becomes

$$\begin{aligned} \frac{Q}{\theta} &= \frac{A_1 \Delta t}{\frac{1}{h_1} + \frac{A_1 L}{A_m k} + \frac{A_1}{h_2 A_2}} = \frac{A_m \Delta t}{\frac{A_m}{h_1 A_1} + \frac{L}{k} + \frac{A_m}{h_2 A_2}} \\ &= \frac{A_2 \Delta t}{\frac{A_2}{h_1 A_1} + \frac{A_2 L}{A_m k} + \frac{1}{h_2}} \end{aligned} \quad (2.7)$$

$$= U_1 A_1 \Delta t = U_m A_m \Delta t = U_2 A_2 \Delta t \quad (2.8)$$

where

$$U_1 = \frac{1}{\frac{1}{h_1} + \frac{A_1 L}{A_m k} + \frac{A_1}{h_2 A_2}} = \text{overall coefficient based on inside area, B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}) \quad (2.9)$$

$$U_m = \frac{1}{\frac{A_m}{h_1 A_1} + \frac{L}{k} + \frac{A_m}{h_2 A_2}} = \text{overall coefficient based on mean area, B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}) \quad (2.10)$$

$$U_2 = \frac{1}{\frac{A_2}{h_1 A_1} + \frac{A_2 L}{A_m k} + \frac{1}{h_2}} = \text{overall coefficient based on outside area, B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}) \quad (2.11)$$

Mean Temperature Difference. In most cases of heat transfer, the temperature difference between the warmer and cooler fluids is not constant but varies throughout the length of the apparatus; therefore, a mean value of temperature difference Δt_m must be used in the heat transfer equations.

For the usual type of parallel or countercurrent operation, or where a flowing fluid is being heated by condensing vapors, the mean temperature difference is taken as the logarithmic-mean of the terminal temperature differences, Δt_1 and Δt_2 :

$$\Delta t_m = \frac{\Delta t_2 - \Delta t_1}{2.303 \log_{10} \frac{\Delta t_2}{\Delta t_1}} \quad (2.12)$$

As Δt_2 approaches Δt_1 , the logarithmic-mean temperature difference approaches the arithmetic mean; and when the ratio $\Delta t_2/\Delta t_1$ is less than 2 it is usually satisfactory to use the arithmetic mean.

It should be realized, however, that in addition to the requirement of either parallel or countercurrent flow, the log-mean temperature difference is based on the assumptions that: (1) there is no change in phase, (2) the mass flow rates are constant, (3) the specific heats are constant, (4) the heat losses are negligible, and (5) the overall heat-transfer coefficient is constant.

If a change of phase occurs, for example where the fluid is being heated and vaporized in the same equipment, a plot of the temperature drop against exchanger length will have a sharp break where vaporization occurs; in this case, the log mean temperature difference has no significance.

Although the assumptions of constant flow rates, constant specific heats, and negligible heat loss are usually not too far wrong, the assumption of a constant overall coefficient is frequently in error, for as the temperature varies throughout the apparatus the viscosity changes, and this has a great effect on the coefficient. However, the coefficient sometimes varies approximately linearly with the temperature drop, and in this case the following relationship applies¹:

$$\frac{Q}{\theta} = A \times \frac{U_1 \Delta t_2 - U_2 \Delta t_1}{2.303 \log_{10} \frac{U_1 \Delta t_2}{U_2 \Delta t_1}} \quad (2.13)$$

This equation can be used to calculate the required area of heating surface if the variation of U with Δt is known, but it cannot be used to calculate the overall coefficient from experimental data.

Film Coefficients. The experimental determination of overall coefficients is one of the simplest of laboratory experiments; it requires only the measurement of the heating area, the flow rates, and the temperatures of the fluid streams at the terminal conditions. The ease with which such experiments can be carried out undoubtedly accounts for their popularity. Unfortunately, however, the results are of limited value, for they can only be applied to the same, or very similar, equipment under similar conditions of operation. The better practice, wherever possible, is to pay more attention to the individual film coefficients.

The determination of film coefficients involves the measurement of surface temperatures, usually by means of thermocouples installed in the tube wall. Such installations must be made with great care if accurate results are to be obtained.

Tube-Wall Thermocouples. The simplest method of installation⁷ is to solder the thermocouples in grooves in the outer surface of the tube: a short groove is cut at right angles to the tube length, one end of a piece of 24-gage copper wire is soldered in the groove, and the excess solder is then filed away to make a continuous smooth surface. A similar groove is cut in the opposite side of the tube, and the end of a piece of 24-gage constantan wire is installed in the same manner. The thermocouple wires are led away vertically from the pipe. Care must be taken to scrape the wires clean to the pipe surface, for any adhering solder causes the thermocouple junction to occur a small distance up the wire, with the result that abnormally high or low readings are obtained.

Colburn and Hougen³ have pointed out that the installation of thermocouples in this manner does not give accurate results because the leads are brought out through a medium that is either hotter or colder than the junction and heat is conducted along the wires. For accurate results, the leads should be brought out through a substantially isothermal zone, preferably through the metal wall itself.

Hebbard and Badger⁶ developed a method in which the thermocouple is soldered in a hole drilled through a chord of the tube wall, and the leads are carried around the tube in a annular groove and withdrawn at a point 180° from the thermocouple junction. The hole and groove are finally filled with a Bakelite cement.

Patton and Feagan⁹ have recently described a method, shown in Figure 1, which has given satisfactory results. A slot is milled in the outside of the tube wall, parallel to the longitudinal axis, from the point at which the junction is to be installed to a point where the leads may be brought out without influencing the heat or fluid flow. A hole, 0.062 inch in diameter, is drilled to a depth of 0.25 inch into the tube wall at an angle from the slot, and the thermocouple junction is inserted in the hole and soldered in place. The thermocouple leads are enclosed in a small brass tube and laid in the slot; this tube is bent slightly at the end and butted against the wall of the slot in such a way that it completely covers the leads. The slot is then filled with solder which is polished down to the original contour of the tube.*

The size of the slot depends on the size of the brass tubing necessary to enclose the leads. For a pair of 24-gage enameled, silk-covered thermocouple wires, a brass tube 0.093 inch in outside diameter and 0.071 inch inside diameter is satisfactory. This size tube fits well in a milled slot about 0.10 inch wide and 0.10 inch deep.

* If the soldered surface is found to influence the flow of condensate over the surface of the tube, the surface may be plated after the thermocouples are installed.

Film-Coefficient Equations. Among the most important factors which determine the magnitude of the film coefficient of heat transfer are the mass velocity of the fluid, the direction of its flow with respect to the surface through which the heat is passing, and whether the flow is viscous or turbulent. Numerous film-coefficient equations have been proposed for the many conditions encountered commercially, but many of these apply only under very limited conditions. Some of the more commonly used equations are:

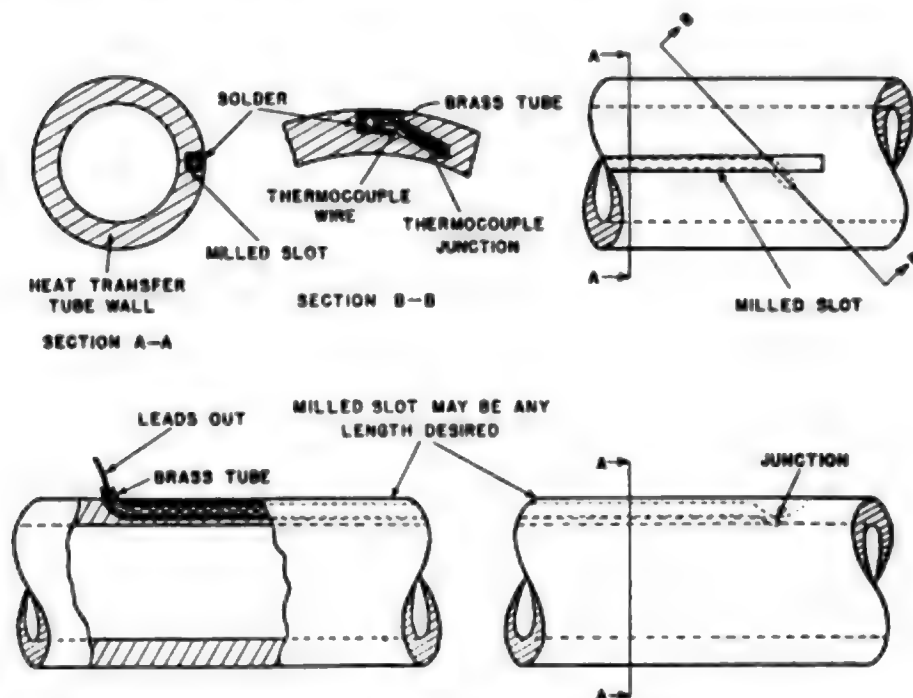


Figure 1. Detail of Installation of Tube-wall Thermocouples by the Patton-Feagan Method

1. For fluids of low viscosity flowing in forced convection inside tubes:
 - a. The Colburn equation,³ which is written as

$$\left(\frac{h}{c_p G}\right) \left(\frac{c_p \mu_f}{k}\right)^{2/3} = 0.023 \left(\frac{DG}{\mu_f}\right)^{-0.2} \quad (2.14)$$

or as

$$h = 0.023 \frac{k}{D} \left(\frac{DG}{\mu_f}\right)^{0.8} \left(\frac{c_p \mu_f}{k}\right)^{1/3} \quad (2.15)$$

- b. The Dittus-Boelter equation:⁴

$$h = 0.0225 \frac{k}{D} \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{c_p \mu}{k}\right)^{0.4 \text{ or } 0.3} \quad (2.16)$$

in which the exponent of the *Prandtl number* $\left(\frac{c_p \mu}{k}\right)$ is 0.4 when the heat is flowing from the surface to the fluid, and 0.3 when the heat is flowing from the fluid to the surface.

2. *For gases in forced convection at right angles:*

a. For single pipes, the general equation¹⁰ is

$$\frac{hD}{k_f} = 0.45 + 0.33 \left(\frac{DG}{\mu_f} \right)^{0.56} \quad (2.17)$$

but for gases at moderate temperatures and with values of $\frac{DG}{\mu_f}$ above 1000, the following simplified equation may be used

$$h = 0.37 c_p \frac{G^{0.56}}{d^{0.44}} \quad (2.18)$$

b. For banks of staggered pipes:^{11, 12}

$$h = 0.33 c_p G_{\max} \left(\frac{DG_{\max}}{\mu_f} \right)^{-0.4} \left(\frac{c_p \mu_f}{k} \right)^{-2/3} \quad (2.19)$$

This equation can also be used for liquids if the Reynolds' number exceeds 2000.

3. *For fluids in natural convection outside of horizontal cylinders:*

a. An approximate general equation is¹³

$$h = 0.471 \frac{k_f}{D} \left(\frac{D^3 \rho_f^2 g_L \beta \Delta t}{\mu_f^2} \right)^{0.26} \left(\frac{c_p \mu_f}{k_f} \right)^{0.25} \quad (2.20)$$

b. For the special case of heat loss by conduction and natural convection from horizontal pipes to air¹⁵

$$h = 0.42 \left(\frac{\Delta t}{d} \right)^{0.25} \quad (2.21)$$

4. *For condensing vapors:*

a. Film-type condensation on horizontal tubes:¹⁴

$$h = 0.73 \frac{k_f}{D} \left(\frac{D^3 \rho_f^2 g_L \lambda}{k_f \mu_f n \Delta t_m} \right)^{0.25} \quad (2.22)$$

b. Film-type condensation on vertical tubes:¹⁴

$$h = 0.94 \frac{k_f}{N} \left(\frac{N^3 \rho_f^2 g_L \lambda}{k_f \mu_f \Delta t_m} \right)^{0.25} \quad (2.23)$$

In these equations

c_p = heat capacity, B.t.u./($\text{lb.} \times ^\circ\text{F.}$).

d = diameter, in.

D = diameter, ft.

g_L = acceleration due to gravity, $4.18 \times 10^8 \text{ ft./hr.}^2$

G = mass velocity, $\text{lb.}/(\text{hr.} \times \text{ft.}^2)$.

G_{\max} = mass velocity based on minimum opening, $\text{lb.}/(\text{hr.} \times \text{ft.}^2)$.

h = film coefficient of heat transfer, $\text{B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.})$.

k = thermal conductivity, $\text{B.t.u.-ft.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.})$.

n = number of rows in a vertical plane.

N = length of heat transfer surface, ft.

Δt_m = mean temperature difference, $^\circ\text{F.}$

β = volumetric coefficient of thermal expansion, $\text{ft.}^3/(\text{ft.}^3 \times ^\circ\text{F.})$.

λ = latent heat of vaporization, B.t.u./lb.

μ = viscosity, $\text{lb.}/(\text{ft.} \times \text{hr.})$.

ρ = density, $\text{lb.}/\text{ft.}^3$

In the Dittus-Boelter equation—(2.16)—the properties of the fluid, μ , ρ , and k , are evaluated at the temperature of the fluid; in the other equations they are evaluated at the mean film temperature t_f or $t_{f'}$, which are defined as

$$t_f = \text{fluid temperature minus } \frac{1}{2}\Delta t_m \quad (2.24)$$

$$t_{f'} = \text{saturation temp. of condensate minus } \frac{1}{2}\Delta t_m \quad (2.25)$$

Heat Transfer in Annular Spaces. When heat is being transferred to fluids in forced convection in ducts that are other than circular in cross section, it is customary to calculate film coefficients by means of either the Colburn or the Dittus-Boelter equation—(2.15) or (2.16)—using in place of the diameter a so-called *equivalent diameter*, which is defined as

$$\begin{aligned} D_{eq} &= 4 \times \text{hydraulic radius} \\ &= 4 \times \frac{\text{cross-sectional area}}{\text{wetted perimeter}} \end{aligned} \quad (2.26)$$

which, for concentric circular annuli becomes

$$D_{eq} = \frac{4 \times (\pi/4)(D_2^2 - D_1^2)}{\pi(D_2 + D_1)} = D_2 - D_1 \quad (2.27)$$

where

D_2 = outer diameter of the annulus.

D_1 = inner diameter of the annulus.

However, the use of an equivalent diameter defined in this manner

frequently leads to erroneous results, particularly for heat transfer through the inner annulus surface.

Monrad and Pelton⁶ have recently found that the use of the equivalent diameter in the Colburn and Dittus-Boelter equations is satisfactory for heat transfer through the outer annulus surface, but that for transfer through the inner surface the use of the equivalent diameter should be accompanied by a change in the constant.

Experimentally, they found that the constant should be changed from 0.023 to 0.020 $\left(\frac{D_2}{D_1}\right)^{0.48}$ but, for general use, they recommended that the constant be calculated from the equation

$$\text{Constant} = 0.023 \left[\frac{2 \log_e \frac{D_2}{D_1} - \left(\frac{D_2}{D_1}\right)^2 + 1}{\frac{D_2}{D_1} - \frac{D_1}{D_2} - 2 \left(\frac{D_2}{D_1}\right) \log_e \frac{D_2}{D_1}} \right] \quad (2.28)$$

The constant calculated from this equation differs considerably from that which has been reported by other investigators.⁸

HEAT TRANSFER BY RADIATION

The net transfer of heat by radiation between two bodies is expressed by the Stefan-Boltzmann law, which is ordinarily written as

$$q_r = 0.173 pA \left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4 \right] \quad (2.29)$$

where

q_r = net heat transferred, B.t.u./hr.

p = emissivity (1.0 for a perfect black body).

A = area of the radiating or receiving surface, ft.²

T_1 = temperature of the hotter body, °R.

T_2 = temperature of the colder body, °R.

In laboratory work, radiation most frequently enters into two types of problems: (1) the calculation of the heat loss from bare and lagged surfaces, and (2) the estimation of radiation errors in pyrometry.

Heat Loss from Bare and Lagged Surfaces. Heat is lost from a pipe, or other surface, to the room in two ways: (1) by conduction through an air film, and then by convection in the bulk of the air; and (2) by direct radiation to the cooler walls of the room.

That portion of the heat which is lost by conduction and convection may be calculated by the equation

$$q_c = h_c A (t_s - t_a) \quad (2.30)$$

and that portion which is lost by radiation may be expressed by the simplified equation

$$q_r = h_r A (t_s - t_w) \quad (2.31)$$

In these equations

q_c = heat transferred from surface to room by conduction and convection, B.t.u./hr.

q_r = heat transferred from surface to room by radiation, B.t.u./hr.

h_c = coefficient of heat transfer by conduction and convection, B.t.u./
(hr. \times ft.² \times °F.).

h_r = coefficient of heat transfer by radiation, B.t.u./(hr. \times ft.² \times °F.).

A = surface area, ft.²

t_s = temperature of the surface, °F.

t_a = temperature of the air, °F.

t_w = temperature of the walls of the room, °F.

The convection coefficient h_c can be evaluated from Equation (2.21):

$$h_c = 0.42 \left(\frac{\Delta t}{d} \right)^{0.25}$$

and the radiation coefficient h_r can be calculated by equating the values of q_r in Equations (2.31) and (2.29):

$$q_r = h_r A (t_s - t_w) = 0.173 p A \left[\left(\frac{T_s}{100} \right)^4 - \left(\frac{T_w}{100} \right)^4 \right]$$

or

$$h_r = \frac{0.173 p \left[\left(\frac{T_s}{100} \right)^4 - \left(\frac{T_w}{100} \right)^4 \right]}{t_s - t_w} \quad (2.32)$$

It is usually more convenient, however, to treat the quantity $(h_c + h_r)$ as if it were a single coefficient, for if the walls of the room are at the same temperature as the air, $t_a = t_w$, and Equations (2.30) and (2.31) can be added to give

$$q = q_c + q_r = (h_c + h_r)(A)(t_s - t_w) \quad (2.33)$$

which may be written as

$$q = (h_c + h_r) A \Delta t \quad (2.34)$$

where

q = total heat lost from the surface by conduction and convection, and by radiation, B.t.u./hr.

$h_c + h_r$ = combined coefficient of heat transfer by conduction and convection, and radiation, B.t.u./(hr. \times ft.² \times °F.).

A = surface area, ft.²

Δt = temperature difference, surface to room, °F.

The combined coefficient ($h_c + h_r$) may be evaluated from the curves of Figure 2, which are based on experimental data for single, hori-

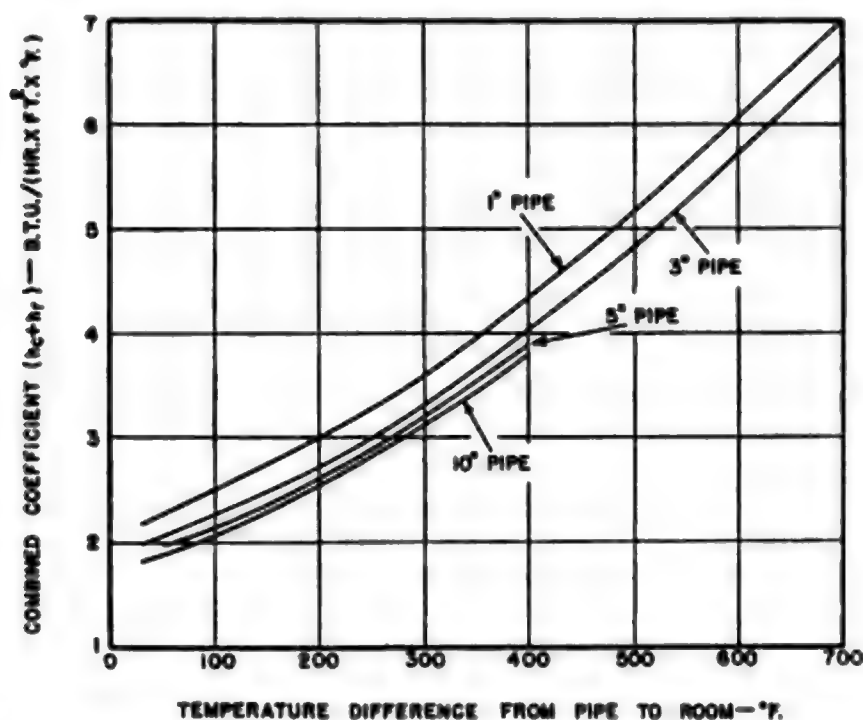


Figure 2. Combined Coefficient of Conduction and Convection, and Radiation for Horizontal Bare Steel Pipe with Room at 80°F.

zontal, bare steel pipes with an oxidized surface.¹⁶ These curves can also be used with reasonable accuracy for ordinary lagged surfaces.

Radiation Errors in Pyrometry. When a thermometer or thermocouple is used to measure the temperature of a gas, the reading will be in error if the temperature of the surrounding walls is different than the temperature of the gas. If the walls are at a higher temperature than the gas, heat will be transferred to the instrument by radiation and from the instrument to the gas by conduction and convection; therefore, the instrument will read too high. If the walls are at a lower temperature than the gas, heat will be transferred from the gas to the instrument

to the walls, and the reading will be too low. In either case, the relationship between the temperatures of the gas, the wall, and the instrument may be expressed as

$$\frac{q}{A} = 0.173 p \left[\left(\frac{T_w}{100} \right)^4 - \left(\frac{T_s}{100} \right)^4 \right] = h_c (T_s - T_g) \quad (2.35)$$

where

$\frac{q}{A}$ = heat transferred by radiation, and by conduction and convection between the gas, the instrument, and the walls, after a steady state has been reached, B.t.u./(hr. \times ft.²).

p = emissivity of the instrument surface (approximately 0.96 for mercury thermometers and 0.90 for thermocouples).

T_w = temperature of the walls, °R.

T_s = temperature reading of the instrument, °R.

T_g = temperature of the gas, °R.

h_c = coefficient of heat transfer by conduction and convection, B.t.u./(hr. \times ft.² \times °F.).

The value of h_c can be estimated from a film-coefficient equation for the particular type of gas flow involved, and the magnitude of the error in the temperature measurement can be calculated by means of Equation (2.35).

In many cases, errors due to radiation are very serious, but they can usually be kept to a minimum by surrounding the thermocouple or thermometer with a bright metal radiation shield.

NOMENCLATURE

A = area, ft.²

c_p = heat capacity, B.t.u./(lb. \times °F.).

d = diameter, in.

D = diameter, ft.

g_L = acceleration due to gravity, 4.18×10^8 ft./hr.²

G = mass velocity, lb./(hr. \times ft.²).

G_{\max} = mass velocity based on minimum opening, lb./(hr. \times ft.²).

h = film coefficient of heat transfer, B.t.u./(hr. \times ft.² \times °F.).

k = thermal conductivity, B.t.u.-ft./(hr. \times ft.² \times °F.).

L = wall thickness, ft.

n = number of rows in a vertical plane.

N = length of heat transfer surface, ft.

p = emissivity.

- q = heat transferred, B.t.u./hr.
 Q = heat transferred, B.t.u.
 R = thermal resistance, ($^{\circ}\text{F.} \times \text{hr.}$)/B.t.u.
 t = temperature, $^{\circ}\text{F.}$
 Δt = temperature difference, $^{\circ}\text{F.}$
 Δt_m = mean temperature difference, $^{\circ}\text{F.}$
 T = temperature, $^{\circ}\text{R.}$
 T_1 = temperature of hotter body, $^{\circ}\text{R.}$
 T_2 = temperature of colder body, $^{\circ}\text{R.}$
 U = overall coefficient of heat transfer, B.t.u./($\text{hr.} \times \text{ft.}^2 \times ^{\circ}\text{F.}$).
 β = volumetric coefficient of thermal expansion, $\text{ft.}^3/(\text{ft.}^3 \times ^{\circ}\text{F.})$.
 θ = time, hr.
 λ = latent heat, of vaporization, B.t.u./lb.
 μ = viscosity, $\text{lb.}/(\text{ft.} \times \text{hr.})$.
 ρ = density, $\text{lb.}/\text{ft.}^3$.

Subscripts

- c = conduction and convection.
 eq = equivalent.
 f or f' = film.
 g = gas.
 m = mean.
 r = radiation.
 s = surface.
 w = wall.
 1 = inner, except in T_1 .
 2 = outer, except in T_2 .

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A DOUBLE-PIPE HEAT EXCHANGER

Designed and Constructed by

The Department of Chemical Engineering
University of New Hampshire

Description. This heat exchanger is made up of four sections of concentric pipes; in the upper two sections heat transfer takes place between water and water, and in the lower two sections it is between steam and water. The water heated in the lower two sections is used as the hot feed for the upper two sections. The piping arrangement of the liquid to liquid sections permits the study of heat transfer in both parallel and countercurrent flow and with hot water in either the inner pipe or the annular space. As shown in Figure 1, the pipes are arranged in a single plane so that the apparatus can be mounted on a wall; thus a minimum of floor space is required.

In addition to thermometers which are provided at the ends of each section, thermocouples are attached to the outer surface of each inner pipe and are also installed in the annulus, so that film coefficients as well as overall coefficients can be determined.

Materials and Cost. The entire apparatus is constructed of standard pipe and fittings with the exception of the inner pipes which are standard brass condenser tubes; these materials are listed in Table 1. The valves are the most expensive part of the system, accounting for nearly 35 per cent of the total material cost of about \$105.

Comments on Construction and Design. 1. Although several different methods can be used to bring out the lead wires of surface thermocouples, the use of flanges has several advantages: (a) the wires are readily insulated by bringing them out through two gaskets between the flanges; (b) the wires can be brought out with a minimum of danger of breaking the thermocouple; (c) the apparatus can be rapidly dismantled if replacement of thermocouples becomes necessary; and (d) thermocouples for measuring the temperature of the liquid in the annular space can be brought out through the same flanges, thereby eliminating the need for special fittings.

Certain precautions, however, must be observed in using flanges: (a) to minimize disturbance to the fluid stream, the flanges should be screwed on so that the end of the pipe is practically flush with the face of the flange; (b) the end of the pipe should be reamed and filed smooth to eliminate burrs; and (c) the gaskets should be cut so that they do not project into the annular space.

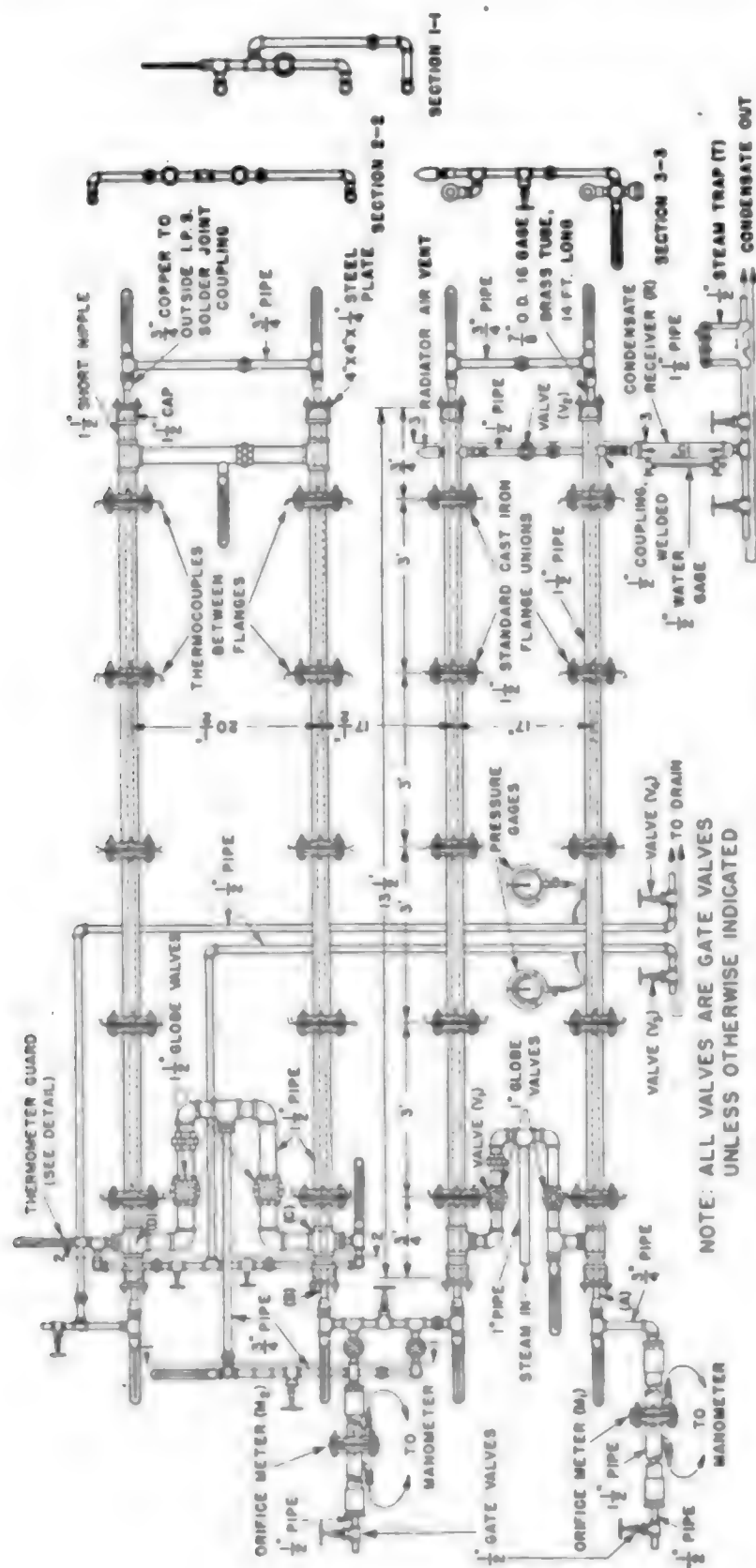


Figure 1. Double-Pipe Heat Exchanger—University of New Hampshire

TABLE 1
Bill of Materials for Double-Pipe Heat Exchanger
 University of New Hampshire

Quantity	Description	Size	Material
63½ ft.	Std. pipe	1½ in.	Black wrought steel
3 ft.	" "	1 in.	" " "
11 ft.	" "	¾ in.	" " "
1 ft.	" "	¾ in.	" " "
1	Std. nipple	1½ in. x 4½ in. long	" " "
1	" "	1½ in. x 3½ in. "	" " "
8	" "	1½ in. x 3 in. "	" " "
1	" "	1½ in. x 2½ in. "	" " "
13	" "	1½ in., short	" " "
5	" "	1½ in., close	" " "
1	" "	1 in. x 10 in. long	" " "
5	" "	1 in. x 6 in. "	" " "
2	" "	1 in. x 4 in. "	" " "
4	" "	1 in. x 3½ in. "	" " "
2	" "	1 in., short	" " "
7	" "	1 in., close	" " "
1	" "	¾ in. x 12 in. long	" " "
1	" "	¾ in. x 9 in. "	" " "
1	" "	¾ in. x 6 in. "	" " "
2	" "	¾ in. x 5 in. "	" " "
2	" "	¾ in. x 4 in. "	" " "
1	" "	¾ in. x 3½ in. "	" " "
1	" "	¾ in. x 3 in. "	" " "
1	" "	¾ in. x 2½ in. "	" " "
6	" "	¾ in. x 2 in. "	" " "
13	" "	¾ in., short	" " "
11	" "	¾ in., close	" " "
4	" "	½ in. x 6 in. long	" " "
1	" "	½ in. x 5 in. "	" " "
2	" "	½ in. x 3½ in. "	" " "
4	" "	½ in. x 3 in. "	" " "
1	" "	½ in. x 2½ in. "	" " "
4	" "	½ in. x 2 in. "	" " "
7	" "	½ in., short	" " "
5	" "	½ in., close	" " "
1	" "	½ in. x 3 in. long	" " "
2	" "	½ in. x 2 in. "	" " "
4	" "	½ in. x 4 in. "	" " "
5	Std. tee	1½ in.	Black malleable iron
2	" "	1 in.	" " "
12	" "	¾ in.	" " "
5	" "	¾ in.	" " "
4	" "	½ in.	" " "

TABLE 1—Continued

Quantity	Description	Size	Material
4	Std. 90° elbow	1½ in.	Black malleable iron
4	" " "	1 in.	" " "
5	" " "	¾ in.	" " "
11	" " "	½ in.	" " "
8	" " "	¼ in.	" " "
2	Ground-joint union	1½ in.	" " "
2	" " "	1 in.	" " "
6	" " "	¾ in.	" " "
3	" " "	½ in.	" " "
4	" " "	¼ in.	" " "
22	Std. flange union, gasket type	1½ in.	Cast iron
3	Bushing	1½ in. x ¾ in.	" "
2	"	1½ in. x ½ in.	" "
1	"	1 in. x ¾ in.	Black malleable iron
2	"	¾ in. x ½ in.	" " "
1	"	½ in. x ¼ in.	" " "
4	Std. coupling	1½ in.	" " "
1	" "	¾ in.	" " "
3	" "	½ in.	" " "
2	Std. reducer	1½ in. x ¾ in.	" " "
2	" "	1½ in. x 1 in.	" " "
1	" "	1 in. x ½ in.	" " "
14	" "	¾ in. x ½ in.	" " "
1	" "	½ in. x ¼ in.	" " "
2	Std. cross	1½ in.	" " "
1	" "	¾ in.	" " "
10	Std. cap	1½ in.	" " "
8	" "	1 in.	" " "
4	Std. gate valve	¾ in.	Brass
10	" " "	½ in.	"
2	" globe "	1½ in.	"
2	" " "	1 in.	"
1	" " "	¾ in.	"
4	Condenser tube	¾ in. O.D. x 14 ft. long x 16-gage	"
2 ft.	Tubing	¾ in.	Copper
35 ft.	"	⅜ in.	"
16	Compression fitting	⅜ in. copper to ½ in. outside I.P.S.	Brass
4	Elbow, compression type	⅜ in. copper to ½ in. outside I.P.S.	"
8	Solder-joint coupling	¾ in. copper to ½ in. outside I.P.S.	"
4	Solder-joint coupling	½ in. copper to ¼ in. outside I.P.S.	"

TABLE 1—*Concluded*

Quantity	Description	Size	Material
3	Gage cock	$\frac{1}{2}$ in. female I.P.S.	Brass
8	Pet cock	$\frac{1}{2}$ in. female I.P.S.	"
4	" "	$\frac{1}{2}$ in. male I.P.S.	"
4	" "	$\frac{1}{2}$ in. male and female I.P.S.	"
2	Round plate	2 $\frac{1}{2}$ in. O.D. x 20 gage	"
3	Pressure gage	0 to 100 lb. press.; 4 in. face	"
1	Steam-gage syphon	$\frac{1}{2}$ in.	Black wrought steel
8	Square plate	3 in. x 3 in. x $\frac{1}{2}$ in.	Steel
$\frac{1}{2}$ sq. ft.	Plate	$\frac{1}{2}$ in. thick	"
32	Bolts and nuts	$\frac{1}{2}$ in. — 16 N.C. x 2 in. long	"
24 ft.	Angles	$\frac{1}{2}$ in. x $\frac{1}{2}$ in. x $\frac{1}{2}$ in.	"
1	Steam trap (125 lbs.)	$\frac{1}{2}$ in.	Cast iron
1	Hoffman No. 4 radiator air-vent	$\frac{1}{2}$ in.	Brass
1	Water gage	$\frac{1}{2}$ in., with 12 in. glass	"
14	Thermometers	-20 to 150°C., in 1°C.	Glass-mercury
20 ft	Tubing	$\frac{1}{2}$ in.	Glass
6 $\frac{1}{2}$ sq. ft.	Gasket material	$\frac{1}{16}$ in. thick	Cranite
650 ft.	Thermocouple wire	24 gage	Copper-constantan
8	Gaskets	1 $\frac{1}{2}$ in. O.D. x $\frac{1}{2}$ in. I.D. x $\frac{1}{2}$ in. thick	Rubber

2. The thermocouples were attached to the tubes by the most commonly used method: by soldering them into longitudinal grooves filed in the surface of the tube and then filing the solder smooth. Although this method is by no means perfect, the tubes were too thin to permit the use of a more elaborate method for attaching the thermocouples.²

3. The various thermometers are not inserted into wells but project directly into the fluid streams; this greatly minimizes the lag between the actual and indicated temperatures, and results in greater accuracy. The thermometers are passed through rubber stoppers which are held in place by the thermometer guards shown in Figure 2.

4. In heat-transfer experiments where steam is the heating medium, the results are often considerably in error when the quantity of condensate is small and it is removed through a steam trap; this is due to the passage of steam through the trap or to the retention of appreciable

quantities of condensate within the system. The use of a condensate receiver, such as *R* in Figure 1, eliminates these difficulties and results in more accurate data.

This receiver is merely a cylinder, made of standard pipe, fitted with a water gage; in use, condensate is drained from the receiver at a rate sufficient to maintain a constant liquid level in the gage glass. The steam trap *T* is used only where it is desired to heat water in the lower sections and not to determine the coefficients.

5. The rates of flow of both streams of water are measured by means of orifice meters inserted in the respective lines. To permit accurate readings over a wide range of rates, each orifice is provided with two

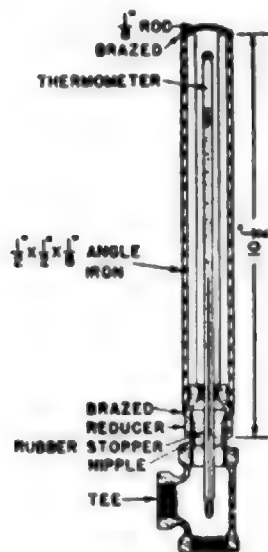


Fig. 2

Figure 2. Detail of Thermometer Guard

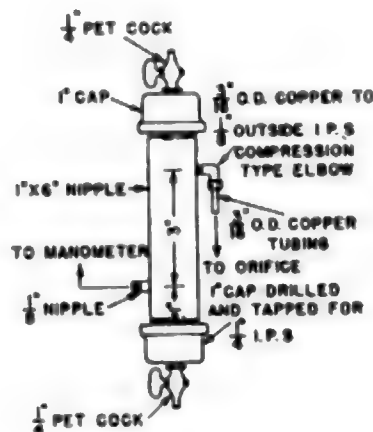


Fig. 3

Figure 3. Detail of Manometer Trap

sets of manometers: one with water over mercury, and the other with water under air. To prevent air and scale from entering the manometers, traps are installed in the lines leading from the orifices. The construction of these traps is shown in Figure 3. Traps of this type can be effectively employed wherever manometers are used.

Operation. During operation, water from the main is passed in parallel through the two orifice meters M_1 and M_2 . The stream through M_1 always enters at point *A* and is heated by steam; it enters the liquid-liquid section at one of three points depending upon the type and condition of flow. The cold water stream likewise enters at one of three points. The entrance points for both streams are as follows:

Type of Liquid Flow	Direction of Heat Flow	Entrance Points	
		Hot Water	Cold Water
Countercurrent	Outside to inside	<i>D</i>	<i>B</i>
	Inside to outside	<i>B</i>	<i>D</i>
Parallel	Outside to inside	<i>C</i>	<i>B</i>
	Inside to outside	<i>B</i>	<i>C</i>

Depending upon the rate of flow and the desired temperature of the hot water, steam may be passed through either one or both of the steam-heated sections; when only one section is used, valves V_1 and V_2 are closed.

The rate of heat transfer in the steam-heated sections is very high, and if the rate of water flow is low the water may be heated to such an extent that it vaporizes in the inner pipe; this would seriously affect the heat transfer measurements in the liquid-liquid sections. To eliminate this difficulty, the discharge valve, V_3 or V_4 , must be partially closed so that the pressure on the system is sufficiently high to prevent vaporization.

TYPICAL EXPERIMENT

Object. To determine the overall and film coefficients of heat transfer of a double-pipe heat exchanger under the following conditions:

1. Parallel flow with hot water in the inner pipe.
2. Countercurrent flow with hot water in the inner pipe.
3. Parallel flow with hot water in the annular space.
4. Countercurrent flow with hot water in the annular space.

Procedure. 1. set the valves for the desired conditions of flow.

2. Turn on the water and adjust the flow of both streams to approximately the desired rates, as indicated by the orifice manometers.

3. Turn on the steam and adjust its pressure so as to obtain a sufficiently high hot water temperature; if the temperature is too high even at the lowest feasible steam pressure, cut out one of the steam-heated sections from the system. If the water is to be heated to a temperature in excess of 212°F. , partially close the discharge valve so that the pressure on the system is greater than the vapor pressure of water at the maximum temperature.

4. Read the various thermometers and thermocouples occasionally until they indicate that equilibrium has been attained, and then start a timed run.

5. At the beginning of the timed run, mark the level of the water in the condensate receiver and begin collecting the condensate in a tared receiver; to prevent loss of condensate through flashing, pass it through a cooler before collecting it. A coil of copper tubing in a tank of water makes a satisfactory cooler.

During the timed run, take several sets of readings of: (a) the steam pressure gage, (b) all thermometer and thermocouples, and (c) the orifice manometers.

At the end of the timed run, make sure that the level in the condensate receiver is at its original value; divert the flow of condensate to the sewer; and weigh the condensate collected.

6. Repeat the experiment under the same conditions of flow but with different water rates.

7. Repeat the experiment for each of the other conditions of flow, making two runs at different rates of flow for each type of flow.

8. From the experimental data, calculate for each run:

- a. The overall heat-transfer coefficient of each of the steam-heated sections.
- b. The overall coefficient of each of the liquid-liquid sections.
- c. The mean outer-film coefficient of each section.
- d. The outer-film coefficient of each section at various points along the exchanger.
- e. The inner-film coefficients from the overall coefficients and the outer-film coefficients.

9. Calculate the mean outer- and inner-film coefficients for each run by means of the:

- a. Dittus-Boelter equation.
- b. Colburn equation.
- c. Modified Colburn equation for water in turbulent motion.

10. Calculate the overall coefficients for each run from the film coefficients computed under 9.

TYPICAL RESULTS AND CALCULATIONS

Some of the experimental data, calculated results, and calculations of a typical experiment are presented here. Because of the large amount of data and calculations involved, and the fact that the treatment of the data for each section of the exchanger is similar, only the results of the lower of the two liquid-liquid sections are included.

The experimental data for the several runs are given in Table 2, and the measurements of the section are listed in Table 3. As an aid in following the calculations, the locations of the thermometers and

thermocouples are shown in diagrammatic form in Figure 4, and the temperatures are plotted as a function of exchanger length in Figures 5, 6, 7, and 8.

Overall Coefficients Calculated from Experimental Data. A summary of the overall coefficients calculated from the experimental data is given

TABLE 2
Experimental Results for Transfer of Heat in Double-Pipe Heat Exchanger
(Average values)

Run No.	Rate of flow of water, lb./hr.		Water temperature, °F.				Temperature* from thermocouple reading, °F.									
	Hot W_A	Cold W_B	Hot		Cold		t_{w_1}	t_{s_1}	t_{w_2}	t_{s_2}	t_{w_3}	t_{s_3}	t_{w_4}	t_{s_4}	t_{w_5}	t_{s_5}
			in	out	in	out										
			t_1	t_2	t_3	t_4										
							Parallel flow—hot water inside, left to right									
1	1,240	1,215	203.2	154.0	38.1	90.5	41	134	52	136	68	126	76	121	85	122
2	1,530	1,480	191.7	150.6	40.1	84.0	43	128	51	125	64	124	72	121	81	120
			t_1	t_2	t_3	t_4	Countercurrent flow—hot water inside, left to right									
3	1,310	1,365	210.2	165.6	72.0	116.2	106	168	94	153	89	141	76	134	72	130
4	1,340	670	211.5	180.5	101.0	155.7	150	188	139	183	130	172	115	163	106	159
			t_1	t_2	t_3	t_4	Countercurrent flow—hot water outside, right to left									
5	1,255	1,540	166.0	128.3	40.4	68.8	130	100	130	96	144	106	151	109	162	114
6	1,285	786	181.4	148.0	41.5	88.7	150	126	151	125	162	132	169	136	179	144
			t_1	t_2	t_3	t_4	Parallel flow—hot water outside, left to right									
7	1,280	858	216.5	167.4	41.8	108.6	213	179	203	162	195	161	182	159	171	151
8	1,280	1,500	215.0	153.1	39.2	87.6	211	162	197	147	185	144	167	139	156	129

* t_w = water temperature.

t_s = tube surface temperature.

in Table 4. The coefficients were calculated from the equation

$$U_m = \frac{Q/\theta}{A_m(\Delta t_m)} \quad (1)$$

where

Q/θ = heat transferred, B.t.u./hr.

= (lb. water per hr. through inner pipe) \times (temp. change, °F.)
 \times (heat capacity).

A_m = mean area of inner pipe, ft.²

Δt_m = log-mean temperature difference from hot water to cold water, °F.

TABLE 3
Measurements of Lower Liquid-Liquid Section of Heat Exchanger

Item	Symbol
Inner Pipe:	
1. Outside diameter = 0.875 in.....	d_o
2. = 0.0729 ft.....	D_o
3. Inside diameter = 0.745 in.....	d_i
4. = 0.0621 ft.....	D_i
5. Wall thickness = 0.065 in.	
6. = 0.00542 ft.....	L
7. Length of heating section = 13.521 ft.....	
8. Outside surface area = $\pi (0.0729) \times 13.521 = 3.095 \text{ ft.}^2$	A_o
9. Inside surface area = $\pi (0.0621) \times 13.521 = 2.636 \text{ ft.}^2$	A_i
10. Log-mean diameter = $\frac{0.875 - 0.745}{2.303 \log \frac{0.875}{0.745}} = 0.805 \text{ in.}$	d_m
11. = 0.0671 ft.....	D_m
12. Mean area of heating surface = $\pi (0.0671) \times 13.521$ = 2.85 ft.^2	A_m
13. Cross-sectional area = 0.00303 ft.^2	

Outer Pipe:

1. Inside diameter = 1.610 in.
2. = 0.1343 ft.

Annulus:

1. Equivalent diameter = 0.735 in.
2. = 0.06125 ft.
3. Cross-sectional area = 0.00996 ft.^2

Distance from left end of heating surface to thermocouple stations:

Station No.	1	2	3	4	5
Distance, ft.	0.692	3.59	6.675	9.68	12.68

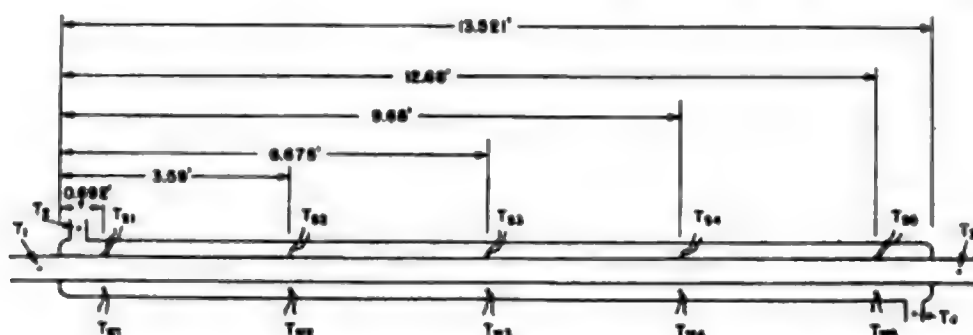


Figure 4. Location of Thermometers and Thermocouples in the Lower Liquid-Liquid Section

FOR RUN NO. 1:

a. $Q/\theta = 1240 \times (203.2 - 154.0) \times 1.0 = 61,000 \text{ B.t.u./hr.}$

b. $A_m = 2.85 \text{ ft.}^2$ (from Table 3).

c. $\Delta t_m = \frac{(\Delta t_l - \Delta t_r)}{2.303 \log_{10} \frac{\Delta t_l}{\Delta t_r}} = \frac{165.1 - 63.5}{2.303 \log_{10} \frac{165.1}{63.5}} = 106.3^\circ\text{F.}$

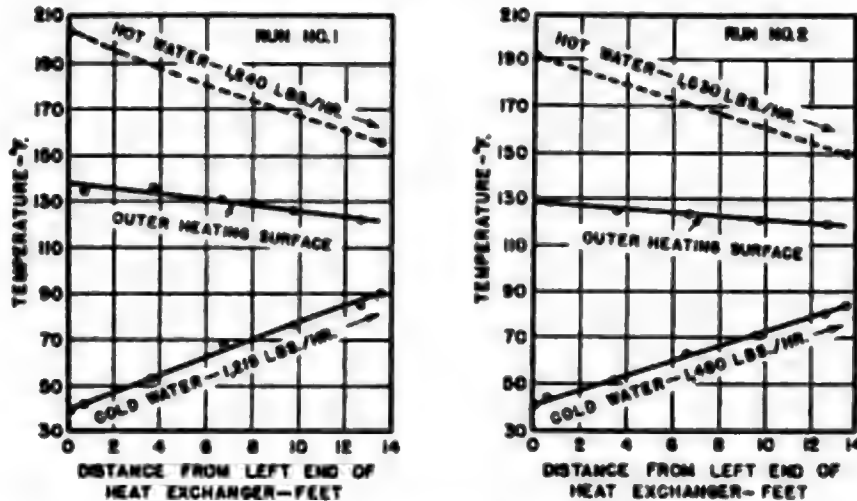


Figure 5. Temperature Readings for Parallel Flow with Hot Water in Inner Pipe

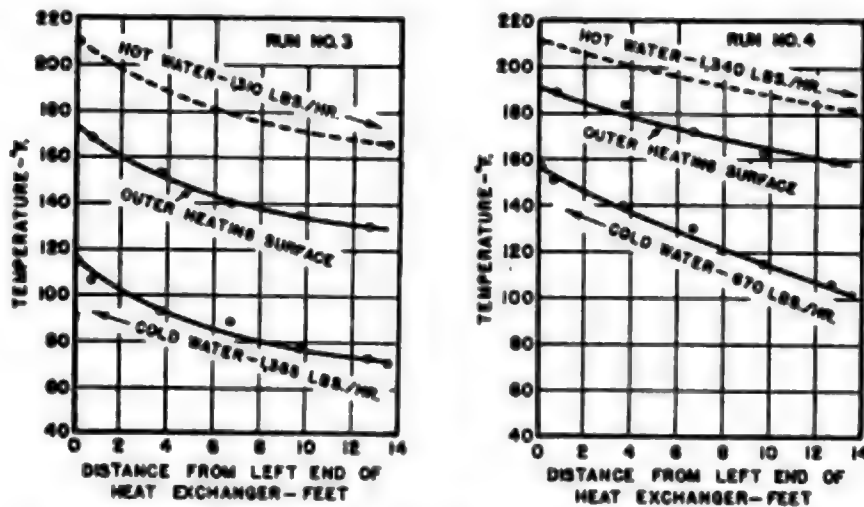


Figure 6. Temperature Readings for Countercurrent Flow with Hot Water in Inner Pipe

where

Δt_l = temp. drop at left end of exchanger
 $= (t_1 - t_2) = 203.2 - 38.1 = 165.1^\circ\text{F.}$ (from Figure 5).

Δt_r = temp. drop at right end of exchanger
 $= (t_3 - t_4) = 154.0 - 90.5 = 63.5^\circ\text{F.}$ (from Figure 5).

Therefore,

$$U_m = \frac{61,000}{(2.85)(106.3)} = 202 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}).$$

Mean Outer-Film Coefficients Calculated from Experimental Data.
The mean outer-film coefficient h_o was calculated from the equation

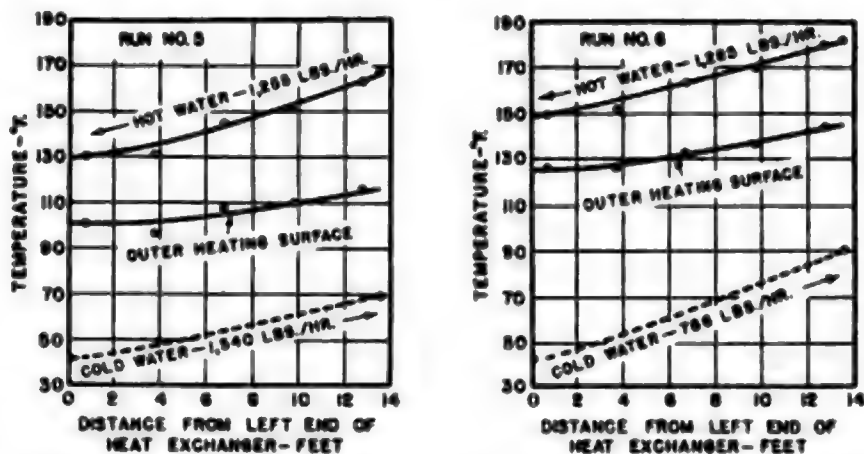


Figure 7. Temperature Readings for Countercurrent Flow with Cold Water in Inner Pipe

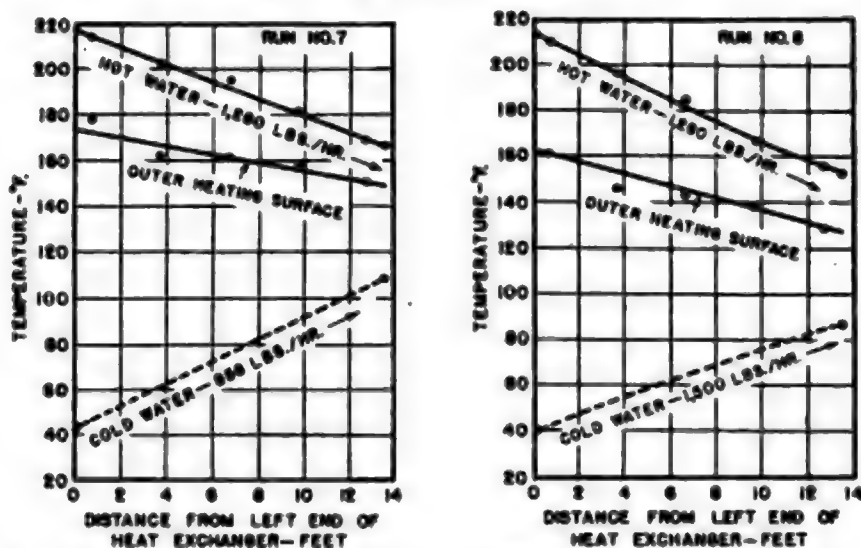


Figure 8. Temperature Readings for Parallel Flow with Cold Water in Inner Pipe

$$h_{om} = \frac{Q/\theta}{A_o(\Delta t_{of})} \quad (2)$$

where

Q/θ = heat transferred, B.t.u./hr.

A_o = outside area of the heating surface, $\text{ft.}^2 = 3.095$ (from Table 3).

Δt_{of} = mean temperature drop, °F. This was obtained as follows: the temperature drops across the outer films were read from Figures 5 to 8 and plotted in Figure 9 as functions of the distance from the left end of the heating surface. The area

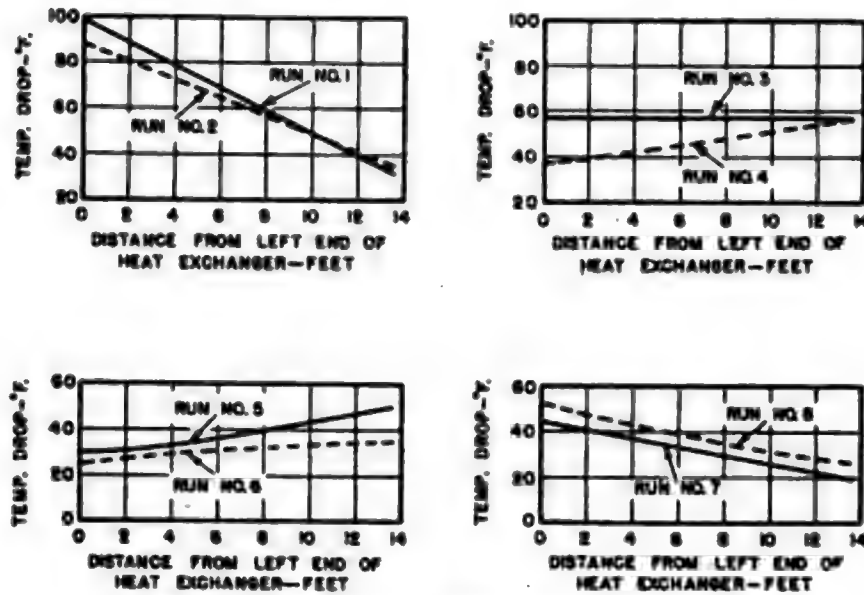


Figure 9. Temperature Drop Across Outer Film

TABLE 4

Summary of Calculations of Experimental Overall- and Outer-Film Coefficients of Heat Transfer

Run No.	Q/θ , heat transferred, B.t.u./hr.	Overall terminal temp. drop, °F.		Δt_m , mean overall temp. drop, °F.	U , overall coefficient, B.t.u./ (hr. \times ft. ² \times °F.)	Δt_{of} , mean outer-film temp. drop, °F.	h_{om} , mean outer-film coefficient, B.t.u./ (hr. \times ft. ² \times °F.)
		t_i	t_r				
1	61,000	165.1	63.5	106.3	202	64.7	305
2	62,900	151.6	66.6	103.2	214	61.5	331
3	58,400	94.0	93.6	93.8	218	56.8	330
4	41,550	55.8	79.5	66.8	218	46.0	292
5	43,700	87.9	97.2	92.2	166	38.1	371
6	37,100	106.5	92.7	98.6	132	31.0	386
7	57,250	174.7	58.8	106.5	189	31.2	594
8	72,700	175.8	65.5	111.6	228	37.8	621

under each curve was determined graphically and divided by the length of the heating surface—13.521 ft.—to obtain Δt_{of} .

The value of Δt_{af} for Run No. 1 was 64.7°F.; therefore,

$$h_{om} = \frac{61,000}{3.095 \times 64.7} = 305 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}).$$

The calculations for all of the runs are summarized in Table 4.

Outer-Film Coefficients at Various Points along the Exchanger. These values were calculated by means of the equation

$$h_o = \frac{W \frac{dt_w}{dN} (c_p)(n)}{\Delta t_o} \quad (3)$$

where

W = rate of flow of water through the annulus, lb./hr.

$\frac{dt_w}{dN}$ = rate of temperature change of the water in the annulus per unit length of heating surface, °F./ft.

n = length of inner pipe per unit of outer surface area, ft./ft.²

Δt_o = temperature drop across the film at any point along the tube, °F. (from Figure 9).

c_p = heat capacity of water = B.t.u./(lb. \times °F.) = 1.0.

These calculations were made in the following manner:

1. The temperature curves of the water in the annulus, as plotted in Figures 5 to 8, were graphically differentiated. The derived curves are plotted in Figure 10.

2. Values of dt_w/dN were read from the curves of Figure 10 for every foot of exchanger length, and corresponding values of Δt_o were read from the curves of Figure 9.

3. The value of n was calculated as follows:

$$n = \frac{\text{length of heating surface}}{A_o} = \frac{13.521}{3.095} = 4.360 \text{ ft./ft.}^2$$

The appropriate values were substituted in Equation (3) and the spot values of the film coefficients along the exchanger length were calculated; the results are plotted in Figure 11, and a summary of the calculations for Run No. 1 is given in Table 5.

Inner-Film Coefficients Calculated from Overall Coefficients and Outer-Film Coefficients. The general equation for heat transfer may be written as

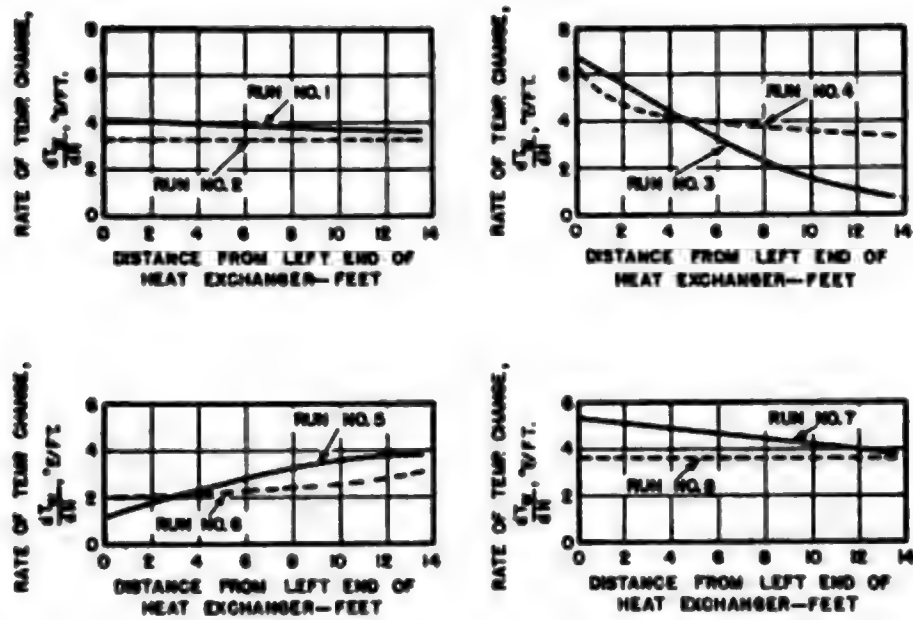


Figure 10. Rate of Change of Temperature of Water in the Annulus

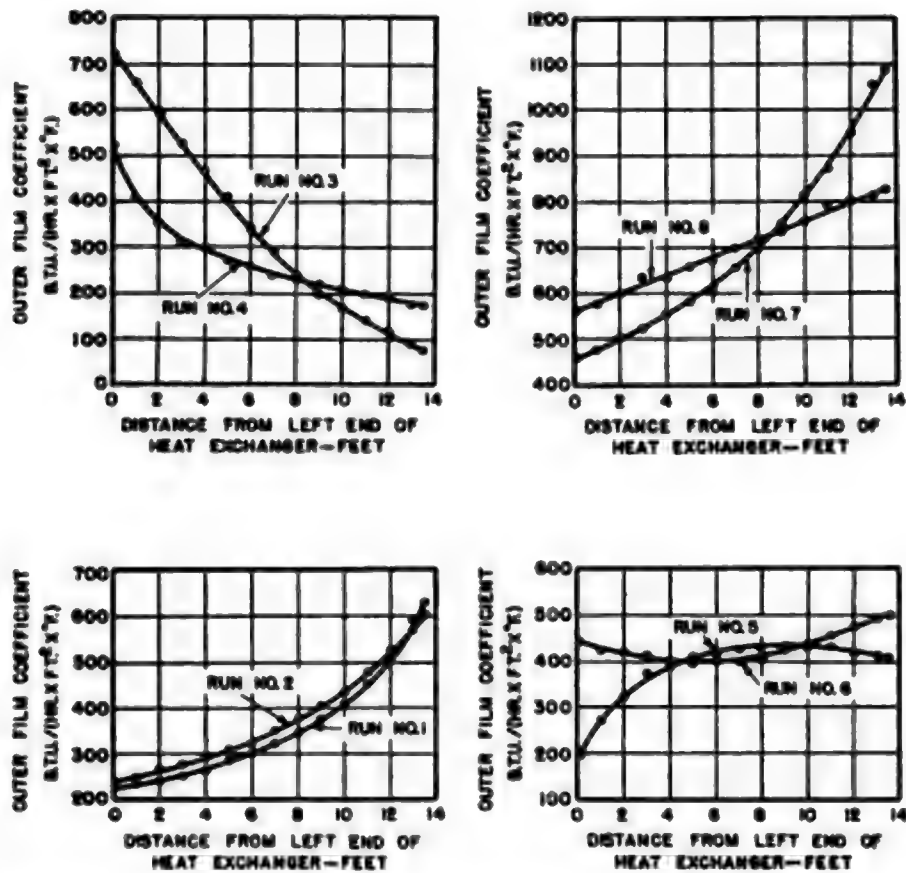


Figure 11. Values of Outer-Film Coefficients Along Exchanger Length

TABLE 5

Summary of Calculations of Outer-Film Coefficient at Various Points along Exchanger Length for Run No. 1

W = rate of flow of water, lb./hr. = 1,215

n = ft. of tube length per ft.² of surface area = 4.360

N , distance from left end of exchanger, ft.	t_w , water temp., °F.	ΔN	Δt_w	$\frac{\Delta t_w}{\Delta N}$	$\frac{dt_w}{dN}$	Δt_o , temp. drop across film, °F.	h_o , outer-film coefficient, B.t.u./hr. \times ft. ² \times °F.)
0	38.1	1	4.1	4.1	4.08	98.1	221
1	42.2	1	4.0	4.0	4.04	93.1	230
2	46.2	1	4.0	4.0	4.00	88.5	240
3	50.2	1	3.8	3.8	3.99	83.5	253
4	54.0	1	4.0	4.0	3.94	78.6	266
5	58.0	1	4.0	4.0	3.90	73.5	281
6	62.0	1	4.0	4.0	3.89	68.8	300
7	66.0	1	3.8	3.8	3.87	63.8	321
8	69.8	1	3.7	3.7	3.81	58.8	343
9	73.5	1	3.6	3.6	3.79	54.1	371
10	77.1	1	3.9	3.9	3.77	49.0	407
11	81.0	1	3.5	3.5	3.71	44.0	447
12	84.5	1	3.6	3.6	3.70	39.1	502
13	88.1	0.521	2.0	3.8	3.69	34.0	575
13.521	90.1				3.67	31.2	624

$$Q/\theta = \frac{\Delta t_m}{\frac{1}{h_{im} A_i} + \frac{L}{k_b A_m} + \frac{1}{h_{om} A_o}} \quad (4)$$

$$= \frac{(\Delta t_m)(A_m)}{\frac{A_m}{h_{im} A_i} + \frac{L}{k_b} + \frac{A_m}{h_{om} A_o}} \quad (5)$$

$$= \frac{(\Delta t_m)(A_m)}{\frac{d_m}{h_{im} d_i} + \frac{L}{k_b} + \frac{d_m}{h_{om} d_o}} \quad (6)$$

$$= U_m A_m (\Delta t_m) \quad (7)$$

where

$$U_m = \frac{1}{\frac{d_m}{h_{im} d_i} + \frac{L}{k_b} + \frac{d_m}{h_{om} d_o}} \quad (8)$$

In these equations

Q/θ = heat transferred, B.t.u./hr.

Δt_m = log-mean overall temperature drop, °F.

A_m = mean heating surface, ft.²

h_{im} = mean inner-film coefficient, B.t.u./(hr. \times ft.² \times °F.).

h_{om} = mean outer-film coefficient, B.t.u./(hr. \times ft.² \times °F.).

L = tube-wall thickness, ft. = 0.00542 (from Table 3).

k_b = thermal conductivity of the metal wall, B.t.u.-ft./(hr. \times ft.² \times °F.) = 69.

A_i = inner heating surface, ft.²

A_o = outer heating surface, ft.²

d_m = mean tube diameter, in. = 0.805 (from Table 3).

d_i = inner tube diameter, in. = 0.745 (from Table 3).

d_o = outer tube diameter, in. = 0.875 (from Table 3).

U_m = overall heat transfer coefficient based on the mean area, B.t.u./(hr. \times ft.² \times °F.).

Substituting the appropriate values in Equation (8):

$$U_m = \frac{1}{\frac{0.805}{h_{im} \times 0.745} + \frac{0.00542}{69} + \frac{0.805}{h_{om} \times 0.875}} \quad (9)$$

$$= \frac{1}{\frac{1.080}{h_{im}} + 0.0000786 + \frac{0.920}{h_{om}}} \quad (10)$$

and

$$\frac{1}{U_m} = \frac{1.080}{h_{im}} + 0.0000786 + \frac{0.920}{h_{om}} \quad (11)$$

and

$$h_{im} = \frac{1.080}{\frac{1}{U_m} - 0.0000786 - \frac{0.920}{h_{om}}} \quad (12)$$

A summary of the calculations of the inner-film coefficients h_{im} by means of Equation (12) is given in Table 6.

Film Coefficients Calculated by Means of Equations. The inner and outer film coefficients for each run were calculated by means of the following three equations:

1. *Dittus-Boelter Equation*¹:

$$h = 0.0225 \left(\frac{k}{D} \right) \left(\frac{DG}{\mu} \right)^{0.8} \left(\frac{c_p \mu}{k} \right)^{0.4 \text{ or } 0.3} \quad (13)$$

TABLE 6

Summary of Calculations of Inner-Film Coefficients from Experimental Overall and Outer-Film Coefficients

Run No.	U_m	h_{om}	$1/U_m$	$\frac{0.920}{h_{om}}$	$\frac{1/U_m - 0.000786}{0.920 - \frac{1}{h_{om}}}$	h_{im}
1	202	305	0.005021	0.003021	0.001920	562
2	214	331	0.004670	0.002783	0.001808	594
3	218	330	0.004577	0.002787	0.001711	630
4	218	292	0.004582	0.003148	0.001353	798
5	166	371	0.006010	0.002478	0.003453	312
6	132	386	0.007571	0.002383	0.005109	211
7	189	594	0.005295	0.001500	0.003716	291
8	228	621	0.004376	0.001481	0.002816	383

where

h = film coefficient, B.t.u./ (hr. \times ft.² \times °F.).

k = thermal conductivity of the fluid, B.t.u.-ft./ (hr. \times ft.² \times °F.).

D = diameter of the channel, ft.

μ = viscosity of the fluid.

G = mass velocity.

c_p = heat capacity of the fluid.

In this equation, G , μ , and c_p must be in consistent units so that the Reynolds' number DG/μ and the Prandtl number $c_p \mu/k$ will be dimensionless. The exponent of the Prandtl number is 0.4 when the fluid is being heated and 0.3 when it is being cooled.

2. *Colburn Equation*:^{3, 5}

$$h = \frac{0.023 c_p G \left(\frac{DG}{\mu_f} \right)^{-0.2}}{\left(\frac{c_p \mu_f}{k} \right)^{1/3}} \quad (14)$$

which may be written as

$$h = 0.023 \left(\frac{k}{D} \right) \left(\frac{DG}{\mu_f} \right)^{0.8} \left(\frac{c_p \mu_f}{k} \right)^{1/3} \quad (15)$$

The symbols for this equation have the same significance and units as in the Dittus-Boelter equation with the exception that the properties of the fluid are evaluated at the mean film temperature instead of at the temperature of the main body of the fluid. The mean film temperature is taken as the arithmetic average of the wall temperature and the fluid temperature.

3. *Modified Colburn Equation:*⁴ A modification of the Colburn equation for use with water in turbulent flow is given as

$$h = \frac{160(1 + 0.012t_f)(V_w)^{0.8}}{(d)^{0.2}} \quad (16)$$

where

t_f = mean temperature of the film, °F.

V_w = velocity of the water, ft./sec.

d = diameter of the pipe, in.

When using any of these equations for a fluid flowing through an annulus, difficulties are encountered in the selection of an appropriate value for the diameter. The usual procedure is to use the *equivalent diameter* defined by the equation

$$\begin{aligned} D &= \frac{4 \times \text{cross-sectional area of annulus}}{\text{wetted perimeter}} \\ &= \frac{4(d_1^2 - d_2^2)\pi}{4(d_1 + d_2)\pi} = d_1 - d_2 \end{aligned} \quad (17)$$

where

d_1 and d_2 are the outer and inner diameters of the annulus, respectively.

The annulus for this exchanger has the following dimensions:

d_1 = inside diameter of 1½-inch pipe = 1.610 in.

d_2 = outside diameter of inner pipe = 0.875 in.

Therefore,

$$D = \frac{1.610 - 0.875}{12} = 0.06125 \text{ ft.}$$

Determination of Mean Temperatures. The mean values of the liquid and film temperatures were determined in the following manner:

1. *Mean temperature of the water in the annulus, t_{aw} .* This was estimated or calculated for each run from the appropriate curve of Figures 5 to 8. In those cases where the temperature curve of the water in the annulus was sufficiently close to a straight line, the mean value could be estimated with a fair degree of accuracy; where the temperature curve deviated appreciably from a straight line, the curve was graphically integrated and the value of the integral was divided by the length of the heating surface to obtain the mean water temperature.

FOR RUN No. 1,

$$t_{aw} = 64.5^{\circ}\text{F. (estimated from curve).}$$

2. *Mean temperature of the water in the inner pipe, t_{iw} .* The mean water temperatures were estimated from the dotted curves of Figures 5 to 8; these curves were drawn with slopes comparable to the slopes of the temperature curves of the water in the annulus. Although the mean temperatures estimated in this manner may be somewhat in error, they are sufficiently close for evaluating the properties of the water.

FOR RUN No. 1,

$$t_{iw} = 178^{\circ}\text{F. (estimated from curve).}$$

3. *Mean outer-film temperature t_{of} .* This temperature is equal to the mean temperature of the water in the annulus, t_{aw} , plus or minus one-half of the mean temperature drop across the outer film, Δt_{of} .

FOR RUN No. 1,

$$t_{of} = 64.5 + \frac{1}{2} \times 64.7 = 96.85^{\circ}\text{F.}$$

4. *Mean inner-film temperature t_{if} .* This temperature is the arithmetic average of the mean temperature of the water in the inner pipe, t_{iw} , and the temperature of the inner wall, t_{ia} . However, to obtain the inner tube-wall temperature, it is necessary to calculate the temperature drop across the tube wall, Δt_b ; this can be done from the experimental overall coefficient U_m and the thermal resistance of the wall. The relationship is

$$\Delta t_b = \frac{\frac{L}{k} \times \Delta t_m}{\frac{1}{U_m}} = \frac{L}{k} \times U_m \times \Delta t_m \quad (18)$$

$$= 0.0000786 \times U_m \times \Delta t_m \quad (19)$$

The mean inner wall temperature t_{iw} is equal to the mean temperature of the water in the annulus, t_{aw} , plus or minus the sum of the mean temperatures drops across the outer film, Δt_{of} , and the metal wall, Δt_b .

FOR RUN NO. 1,

$$\Delta t_b = 0.0000786 \times 202 \times 106.3 = 1.7^\circ\text{F.}$$

$$t_{iw} = 64.5 + 64.7 + 1.7 = 130.9^\circ\text{F.}$$

$$t_{if} = \frac{1}{2} (130.9 + 178) = 154.5^\circ\text{F.}$$

TABLE 7

Data for Run No. 1 for Calculation of Film Coefficients by Means of Equations

Symbol	Definition	Unit	For outer film	For inner film
W	Water rate	lb./hr.	1,215	1,240
D	Diam. of channel	ft.	0.06125	0.0621
d	Diam. of channel	in.	0.735	0.745
A	Cross-sectional area of channel	ft. ²	0.00996	0.00303
G	Mass velocity = (W/A)	lb./(hr. \times ft. ²)	121,600	409,000
V_w	Velocity of water = $G/(62.3 \times 3600)$	ft./sec.	0.542	1.825
t_w	Mean water temp.	$^\circ\text{F.}$	64.5	178.0
t_f	Mean film temp.	$^\circ\text{F.}$	96.85	154.5
k	Thermal conductivity at mean water temp.	$\frac{\text{B.t.u.} - \text{ft.}}{\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}}$	0.342	0.388
k_f	Thermal conductivity at mean film temp.	$\frac{\text{B.t.u.} - \text{ft.}}{\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}}$	0.360	0.382
μ	Viscosity at mean water temp.	lb./(ft. \times hr.)	2.56	0.847
μ_f	Viscosity at mean film temp.	lb./(ft. \times hr.)	1.719	1.012
c_p	Heat capacity	B.t.u./(lb. \times $^\circ\text{F.}$)	1.0	1.0

Calculation of the Outer- and Inner-Film Coefficients for Run No. 1 by Means of Equations. The data for this run are given in Table 7; the calculations are:

1. *By Dittus-Boelter Equation*

$$h = 0.0225 \frac{k}{D} \left(\frac{DG}{\mu} \right)^{0.8} \left(\frac{c_p \mu}{k} \right)^{0.4 \text{ or } 0.3}$$

$$\begin{aligned}\text{Outer film: } h_o &= 0.0225 \times \frac{0.342}{0.06125} \\ &\quad \times \left(\frac{0.06125 \times 121,600}{2.56} \right)^{0.8} \left(\frac{1.0 \times 2.56}{0.342} \right)^{0.4} \\ &= 166 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}).\end{aligned}$$

$$\begin{aligned}\text{Inner film: } h_i &= 0.0225 \times \frac{0.388}{0.0621} \\ &\quad \times \left(\frac{0.0621 \times 409,000}{0.847} \right)^{0.8} \left(\frac{1.0 \times 0.847}{0.388} \right)^{0.4} \\ &= 674 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}).\end{aligned}$$

2. By Colburn Equation

$$h = 0.023 \frac{k_f}{D} \left(\frac{DG}{\mu_f} \right)^{0.8} \left(\frac{c_p \mu_f}{k_f} \right)^{1/3}$$

$$\begin{aligned}\text{Outer film: } h_o &= 0.023 \times \frac{0.360}{0.06125} \\ &\quad \times \left(\frac{0.06125 \times 121,600}{1.719} \right)^{0.8} \left(\frac{1.0 \times 1.719}{0.360} \right)^{1/3} \\ &= 185 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}).\end{aligned}$$

$$\begin{aligned}\text{Inner film: } h_i &= 0.023 \times \frac{0.382}{0.0621} \\ &\quad \times \left(\frac{0.0621 \times 409,000}{1.012} \right)^{0.8} \left(\frac{1.0 \times 1.012}{0.382} \right)^{1/3} \\ &= 647 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}).\end{aligned}$$

3. By Modified Colburn Equation

$$h = \frac{160(1 + 0.012 t_f)(V_w)^{0.8}}{(d)^{0.2}}$$

$$\begin{aligned}\text{Outer film: } h_o &= \frac{160(1 + 0.012 \times 96.85)(0.542)^{0.8}}{(0.735)^{0.2}} \\ &= 226 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}).\end{aligned}$$

$$\begin{aligned}\text{Inner film: } h_i &= \frac{160(1 + 0.012 \times 154.5)(1.825)^{0.8}}{(0.745)^{0.2}} \\ &= 784 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}).\end{aligned}$$

Calculation of Overall Coefficients for Run No. 1 from Film Coefficients Calculated by Means of Equations. These were calculated from Equation (10):

$$U_m = \frac{1}{\frac{1.080}{h_{im}} + 0.0000786 + \frac{0.920}{h_{om}}}$$

1. *From the Dittus-Boelter Equation Values:*

$$\begin{aligned} U_m &= \frac{1}{\frac{1.080}{674} + 0.0000786 + \frac{0.920}{166}} \\ &= 139 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}). \end{aligned}$$

2. *From the Colburn Equation Values:*

$$\begin{aligned} U_m &= \frac{1}{\frac{1.080}{647} + 0.0000786 + \frac{0.920}{185}} \\ &= 151 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}). \end{aligned}$$

3. *From the Modified Colburn Equation Values:*

$$\begin{aligned} U_m &= \frac{1}{\frac{1.080}{784} + 0.0000786 + \frac{0.920}{226}} \\ &= 181 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}). \end{aligned}$$

A summary of all of the experimental and calculated heat-transfer coefficients is given in Table 8, and the corresponding values of the mass velocity and the Reynolds' number are given in Table 9.

Comments on Results. The results of this experiment show that:

1. The experimental outer-film coefficients are much higher than the values calculated by means of the equations.

2. The experimental inner-film coefficients are somewhat lower than the calculated values but, on the whole, the agreement is better than for the outer film.

3. With the exception of one value, the experimental overall coefficients are somewhat lower than the values calculated by means of the Dittus-Boelter and the Colburn equations, although they agree very well with those calculated by means of the modified Colburn equation.

It is possible that part of the discrepancy between the experimental

and calculated results may be due to errors in the measurement of the surface temperatures. Such errors would most likely be due to heat conductance along the wires; this would cause the surface temperature readings to be either higher or lower than the true values, depending upon whether the heat was flowing from the fluid to the wall or from

TABLE 8
Summary of Experimental and Calculated Heat-Transfer Coefficients,
B.t.u./($\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}$)

Run No.	Outer-Film				Inner-Film				Overall			
	Experimental	Dittus-Boelter	Colburn	Modified Colburn	Experimental	Dittus-Boelter	Colburn	Modified Colburn	Experimental	Dittus-Boelter	Colburn	Modified Colburn
1	305	166	185	226	562	674	647	784	202	139	151	181
2	331	193	213	260	594	776	745	900	214	161	171	208
3	330	209	211	273	630	717	702	847	218	167	168	212
4	292	145	150	182	798	759	770	936	218	128	132	159
5	371	227	211	268	312	480	517	633	166	157	153	192
6	386	254	250	305	211	296	341	414	132	136	145	175
7	594	284	281	339	291	349	408	492	189	156	167	205
8	621	271	267	323	383	503	594	718	228	178	187	225

TABLE 9
Calculated Values of Mass Velocity and Reynolds' Number

Run No.	Mass Velocity, $G = \text{lb.}/(\text{hr.} \times \text{ft.}^2)$		Reynolds' Number, DG/μ			
			Calculated from mean water temp.		Calculated from mean film temp.	
	Inner pipe	Annulus	Inner pipe	Annulus	Inner pipe	Annulus
1	409,000	121,600	30,000	2,910	25,100	4,335
2	504,800	148,600	35,000	3,490	29,370	5,120
3	432,000	137,000	32,500	4,375	28,420	5,435
4	442,000	67,300	35,500	3,267	33,510	3,925
5	508,000	126,000	10,440	6,960	14,910	5,942
6	259,400	129,000	6,259	8,375	9,518	7,410
7	283,200	128,500	7,935	10,160	12,680	9,280
8	494,800	128,500	11,900	9,560	19,400	8,360

the wall to the fluid. In either case, the temperature drop across the film would be too low and the outer-film coefficient would be too high. Furthermore, as the inner-film coefficient is calculated from the overall coefficient and the outer-film coefficient, it would tend to be too low.

However, conductance along the thermocouple wires by no means

accounts for all of the discrepancy between the experimental and the calculated outer-film coefficients; some other, more important, factor must be involved. This factor, most probably, is related to the selection of an equivalent diameter for use in the equations when flow is through the annulus. Although it is customary to use a value of four times the hydraulic radius as the equivalent diameter, there is no real justification for this practice and its use unquestionably leads to erroneous results.

Some of the values of the Reynolds' number, particularly for flow through the annulus, are rather low and are in the region where it might be questioned whether the flow was viscous or turbulent. Obviously, if the flow were viscous, the values of the film coefficient calculated by means of the equations would be meaningless, for these equations apply only to conditions of turbulent flow. However, the high experimental values of the outer-film coefficient indicate that the flow was turbulent. Therefore, the calculated values of the Reynolds' number are probably too low. This would be the case if the value of the equivalent diameter was too small; this would also account for the fact that the calculated outer-film coefficients are lower than the experimental values.

Another factor which has considerable influence on the value of the Reynolds' number is the temperature at which it is evaluated; evaluation at the mean liquid temperature gives appreciably different results than evaluation at the mean film temperature; this is evident from Table 8.

From a practical point of view, there seems to be little choice between the various film equations as to the accuracy of the results; however, the Dittus-Boelter equation has the advantage of not requiring a knowledge of film temperatures, and is, therefore, ordinarily much easier to apply.

NOMENCLATURE

- A = area, ft.²
- c_p = heat capacity of water, B.t.u./($\text{lb.} \times ^\circ\text{F.}$).
- d = diameter, in.
- D = diameter, ft.
- G = mass velocity, $\text{lb.}/(\text{hr.} \times \text{ft.}^2)$.
- h = film coefficient of heat transfer, $\text{B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.})$.
- k = thermal conductivity, $\text{B.t.u.-ft.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.})$.
- L = tube wall thickness, ft.
- n = length of inner pipe per unit of outer surface area, $\text{ft.}/\text{ft.}^2$

N = distance from left end of exchanger, ft.

Q = heat transferred, B.t.u.

t = temperature, °F.

Δt = temperature difference, °F.

U = overall heat transfer coefficient, B.t.u./(hr. \times ft.² \times °F.)

V = velocity ft./sec.

W = flow rate, lb./hr.

μ = viscosity, lb./(hr. \times ft.).

θ = time, hr.

Subscripts

b = brass tube wall.

c = cold.

f = film.

h = hot.

i = inner.

l = left end.

m = mean.

o = outer.

r = right end.

s = tube surface.

w = water.

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A CONCENTRIC-PIPE HEAT EXCHANGER

Constructed by

The Department of Chemical Engineering
University of North Dakota

Description. This heat exchanger, which is similar to the one used by Morris and Whitman,¹ is constructed of standard pipe and fittings and a length of 16-gage, $\frac{1}{2}$ -inch O.D., standard brass condenser tubing. The system (Figure 1) is arranged so that fluid flowing through the brass tube is heated by either steam or hot water in the annular space. When hot water is the heating medium, it can be passed either parallel or countercurrent to the fluid being heated. Thermocouples are provided at the inlets and the outlets of both the inner tube and the outer pipe and also at three points along the surface of the tube.

Materials and Cost. The materials used in the construction of this heat exchanger are listed in Table 1; their cost is approximately \$69. This does not include the following auxiliary equipment which is needed when making an experimental run:

1. Potentiometer: Leeds and Northrup portable No. 8657C, or similar.
2. Platform scale: 500- or 1000-pound capacity.
3. Weighing tank: 55-gallon oil drum.
4. Stop watch.

Notes on Construction. Concentric-Pipe Connection. The inner tube is secured to the outer jacket by a packing plate and a conical rubber gasket, as shown in Detail A of Figure 1. Each end of the tube passes through a hole in a standard 2-inch malleable iron cap. The holes in the cap are drilled slightly larger than the diameter of the tube, and are countersunk to receive the rubber gaskets which are held in place by the packing plates that are drilled and countersunk in the same manner as the caps. The packing plates are made of $\frac{1}{4}$ -inch steel plate, 4 inches in diameter; they are drilled to receive bolts that are brazed to the caps.

Thermocouple Arrangements. Surface temperature thermocouples T_1 , T_2 , and T_3 are attached by soldering them into grooves filed in the outer surface of the brass tube. The solder is then filed flush with the tube surface. The thermocouple wires are passed through the outer jacket and secured by the arrangement shown in Detail B of Figure 1. Holes, approximately $\frac{1}{4}$ -inch in diameter, are drilled in the outer jacket to receive the thermocouple wires, and standard malleable iron coup-

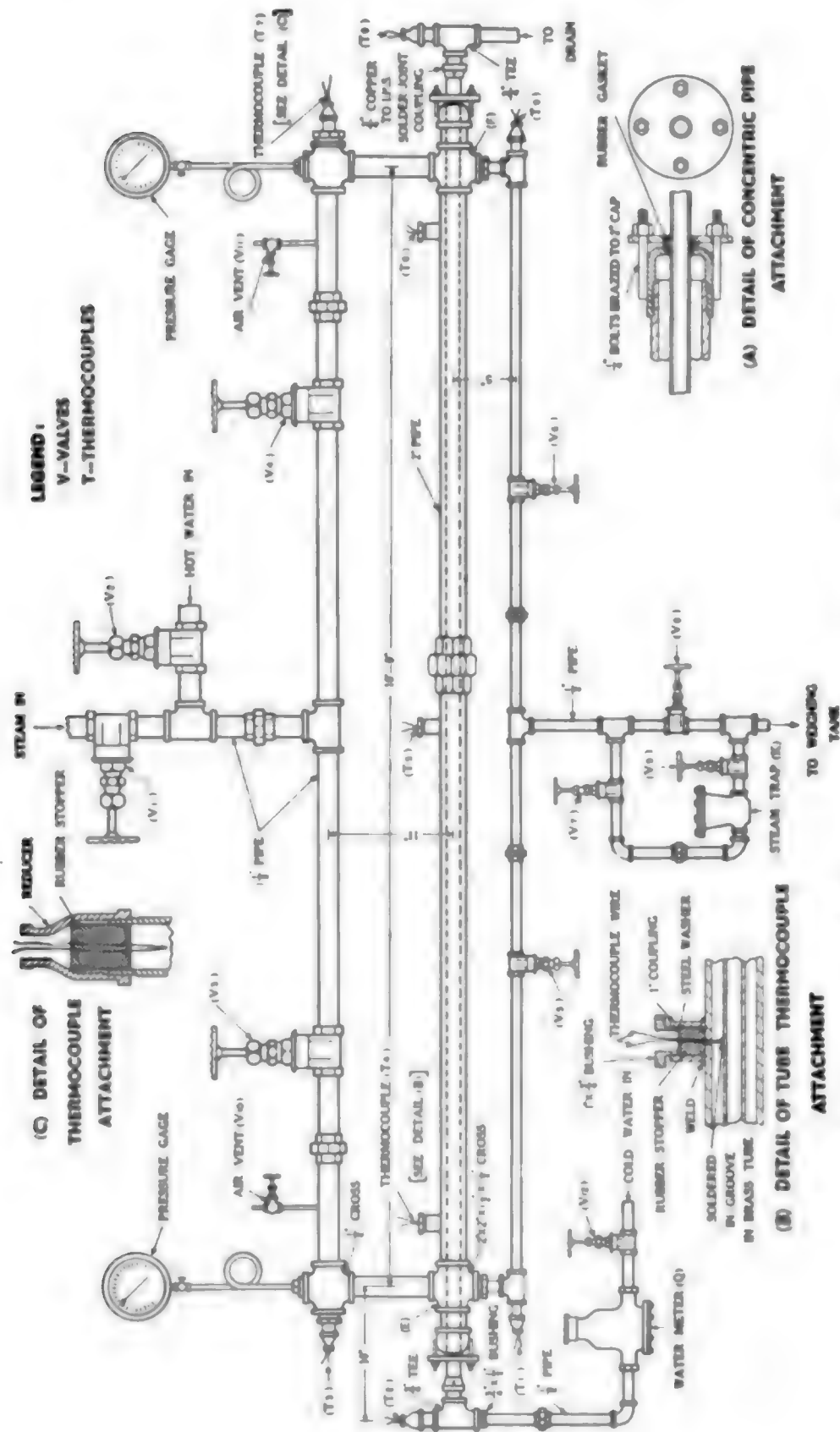


Figure 1. Concentric-Pipe Heat Exchanger—University of North Dakota

TABLE 1
Bill of Materials for Concentric-Pipe Heat Exchanger
 University of North Dakota

Quantity	Description	Size	Material
10.5 ft.	Std. pipe	2 in.	Black wrought steel
10 ft.	" "	1½ in.	" " "
11 ft.	" "	½ in.	" " "
2	Std. nipple	2 in., close	" " "
2	" "	1½ in. x 8 in. long	" " "
4	" "	1½ in. x 6 in. "	" " "
4	" "	¾ in., short	" " "
10	" "	½ in. x 6 in. long	" " "
6	" "	½ in. x 2½ in. "	" " "
1	" "	½ in., short	" " "
2	" "	½ in. x 6 in. long	" " "
2	Std. tee	1½ in.	Black malleable iron
2	" "	¾ in.	" " "
5	" "	½ in.	" " "
3	Std. 90° elbow	½ in.	" " "
1	Ground-joint union	2 in.	" " "
3	" " "	1½ in.	" " "
4	" " "	½ in.	" " "
2	Bushing	1½ in. x ¾ in.	Cast iron
2	"	1½ in. x ½ in.	" "
3	"	1½ in. x ½ in.	" "
1	"	1 in. x ¾ in.	Black malleable iron
4	Std. reducer	¾ in. x ½ in.	" " "
2	" "	½ in. x ½ in.	" " "
3	Std. coupling	1 in.	" " "
2	Std. cap	2 in.	" " "
2	Std. cross	2 x 2 x 1½ x 1½ in.	" " "
2	" "	1½ in.	" " "
4	Std. gate valve	1½ in.	Brass
6	" " "	½ in.	"
2	Std. globe valve	½ in.	"
14 ft.	Std. condenser tube	¾ in. O.D. x 16 gage	"
1	Steam trap, inverted open-float type	½ in.	Cast iron
2	Pressure gage	0 to 100 lb.; 3½-in. face	Brass
2	Steam-gage siphon	½ in.	Black wrought steel
1	Water meter	½ in.	Bronze
2	Round plate	4 in. O.D. x ½ in. thick	Steel
50 ft.	Thermocouple wire	28 gage	Copper-constantan
12 ft.	Pipe insulation	2 in.	85% magnesia
15 ft.	" "	1½ in.	" "

TABLE 1—*Concluded*

Quantity	Description	Size	Material
2	Gasket for brass tube	1½-in. O.D. x ½-in. I.D. x ⅛-in. thick	Rubber
2	Stopper	No. 1	"
4	"	No. 4	"
3	"	No. 7	"
2	Coupling, solder-joint type	½-in. copper to ¾-in. outside I.P.S.	Brass

lings, that have been ground on one end to the contour of the pipe, are welded in place over each hole. A rubber stopper with two small holes is fitted into each coupling, the thermocouple wires are passed through the holes, and the stopper is then compressed by a steel washer and a bushing that screws into the coupling.

Fluid temperature thermocouples T_1 , T_2 , T_3 , T_7 , T_8 , and T_9 are inserted by means of a somewhat simpler arrangement, shown in Detail C. The thermocouple wires are passed through holes in a rubber stopper securely fitted into the end of a nipple that is screwed into a tee. The rubber stopper is held in place by a reducer that is screwed to the nipple.

Use of Thermometers. If desired, thermometers may be used in place of fluid temperature thermocouples. If this is done, it is advisable to protect the thermometers with guards. An excellent guard, which can be made in the laboratory shop, is shown in Figure 2.*

Comments on Design. The Department of Chemical Engineering of *Marquette University*** recommends the following method for making a concentric-pipe connection: a reducing tee with a pipe nipple is attached to the outer pipe, and through this is passed the inner pipe; connection is made by means of a drilled pipe cap, a slip ring, and packing.

Operation. Using Steam. With valves V_2 , V_4 , V_6 , and V_8 closed, steam is passed through valves V_1 and V_3 ; it enters the annular space at E and leaves as condensate at F ; the condensate then passes to the weighing tank through valves V_4 , V_7 , and V_9 and steam trap K . Valve V_{11} is opened slightly to permit the escape of air from the steam space.

Water is introduced through valve V_{12} and water meter Q , passes through the inner tube, and discharges to the drain.

Using Hot Water in Parallel Flow. Hot water is introduced through valve V_2 and follows the same path as described for the flow of steam.

* Through courtesy of R. M. Boarts.

** Private communication, R. W. Ludt.

except that it by-passes the steam trap *K* and passes directly to the weighing tank. The cold water follows the same path as described above.

Using Hot Water in Countercurrent Flow. During this operation, valves V_3 and V_6 are closed and valves V_4 and V_5 are opened. The hot water, therefore, enters the annular space at *F* and leaves at *E*.

TYPICAL EXPERIMENT

Object. To determine the liquid-liquid overall coefficient of heat transfer using hot and cold water in countercurrent flow.

Procedure. 1. Measure the length of the heating section and calculate the area of the heating surface.

2. Study the assembly to become familiar with the method of operation, and then set the valves for countercurrent flow.

3. After conditions of flow and temperature have become constant, record the following readings at 2-minute intervals:

- a. Time.
- b. Inlet and exit temperatures of the cold and the hot water streams.*
- c. Water meter readings.
- d. Weight of discharge.

4. Repeat the experiment using different rates of flow. Each run should last for approximately 20 minutes.

5. Calculate the overall heat-transfer coefficient for each run.

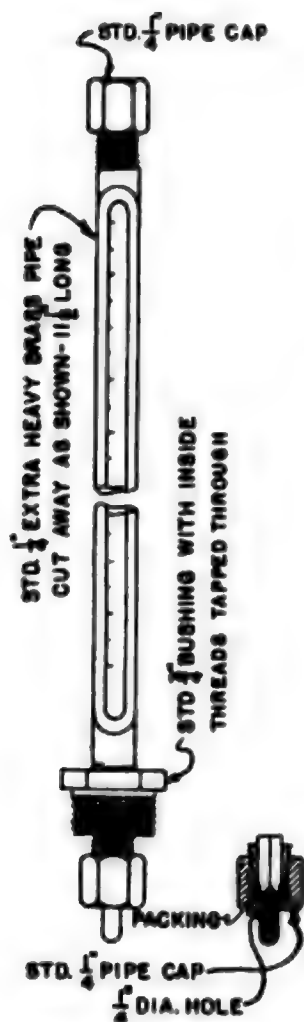


Figure 2. Design of Thermometer Guard—University of Tennessee

Typical Data. A part of the data obtained in an experiment with this heat exchanger is given in Table 2.

Sample Calculations.

Item 1: Average heating surface, $ft.^2$

$$= (\text{length of tube, ft.}) \left(\frac{O.D. + I.D.}{2} \right) \frac{\pi}{12}$$

* Hot water for this experiment was obtained by mixing live steam with cold water in a tank, from which the hot water was pumped to the system.

$$= 11.5 \left(\frac{0.875 + 0.745}{2} \right) \frac{\pi}{12} = 2.44$$

Item 2: *Rate of flow of hot water, lb./hr.*

$$= (\text{lb./min.}) \times 60 = \frac{(460 - 245)}{20} \times 60 = 645$$

TABLE 2

Experimental Heat-Transfer Data for Countercurrent Flow

Length of heating section = 11.5 ft.

Outside diameter of heating tube = 0.875 in.

Inside diameter of heating tube = 0.745 in.

Time, min.	Cold Water			Hot Water		
	Meter reading, gal.	Temp. °F.		Scale reading, lb.	Temp. °F.	
		Inlet	Exit		Inlet	Exit
0	7968.2	48.4	102.2	245	199.5	147.2
2	7970.5	48.3	102.2	267	199.0	146.5
4	7973.0	47.7	102.2	289	199.6	147.2
6	7975.4	48.1	102.2	310	200.4	147.8
8	7977.9	47.6	102.7	332	200.0	148.4
10	7980.3	48.0	101.9	353	199.6	147.2
12	7982.8	48.0	102.0	374	199.5	147.0
14	7985.2	47.5	102.0	396	199.8	147.2
16	7987.7	47.4	102.1	418	199.5	147.6
18	7990.0	47.7	101.9	439	200.0	147.0
20	7992.5	47.9	101.8	460	200.2	147.4
Average.....		47.9	102.1		199.7	147.3

Item 3: *Rate of flow of cold water, lb./hr.*

$$= (\text{gal./min.}) \times 60 \times 8.33$$

$$= \frac{(7992.5 - 7968.2)}{20} \times 60 \times 8.33 = 607$$

Item 4: *Temperature drop of hot water, °F.*

$$= \text{average inlet temp.} - \text{average exit temp.}$$

$$= 199.7 - 147.3 = 52.4$$

Item 5: *Temperature rise of cold water, °F.*

$$= \text{average exit temp.} - \text{average inlet temp.}$$

$$= 102.1 - 47.9 = 54.2$$

Item 6: *Heat given up by hot water, B.t.u./hr.*

$$= \text{Item 2} \times \text{Item 4} = 645 \times 52.4 = 33,800$$

Item 7: *Heat taken up by cold water, B.t.u./hr.*

$$= \text{Item 3} \times \text{Item 5} = 607 \times 54.2 = 32,900$$

Item 8: *Heat loss, B.t.u./hr.*

$$= \text{Item 6} - \text{Item 7} = 33,800 - 32,900 = 900$$

Item 9: *Heat loss, per cent*

$$= (\text{Item 8}/\text{Item 6}) \times 100 = (900/33,800) \times 100 = 2.66$$

Item 10: *Arithmetic-mean temperature difference °F.*

$$= \frac{1}{2}(\Delta t_1 + \Delta t_2) = \frac{1}{2}(97.6 + 99.4) = 98.5$$

where

$$\begin{aligned}\Delta t_1 &= \text{inlet temp. of hot water} - \text{exit temp. of cold water} \\ &= 199.7 - 102.1 = 97.6\end{aligned}$$

$$\begin{aligned}\Delta t_2 &= \text{exit temp. of hot water} - \text{inlet temp. of cold water} \\ &= 147.3 - 47.9 = 99.4\end{aligned}$$

Item 11: *Overall coefficient of heat transfer, B.t.u./(hr. \times ft.² \times °F.)*

$$= q/A\Delta t = 32,900/(2.44 \times 98.5) = 137$$

where

$$q = \text{heat transferred, B.t.u./hr.} = \text{Item 7} = 32,900$$

$$A = \text{average area of heating surface, ft.}^2 = 2.44$$

$$\Delta t = \text{mean temp. difference, °F.} = \text{Item 10} = 98.5$$

REFERENCE

1. MORRIS, F. H., AND WHITMAN, W. G., *Ind. Eng. Chem.*, **20**, 234-40 (1928).

A CONCENTRIC-PIPE HEAT EXCHANGER

Designed and Constructed by
The Department of Chemical Engineering
Drexel Institute of Technology*

Description. This heat exchanger, which is constructed of standard pipe and fittings, consists of four sections of concentric pipes. In one section, water flows through the inner pipe and is heated by steam in the annular space; in the other sections, the water which has been heated passes through the annular spaces and is cooled by water flowing through the inner pipes in either parallel or countercurrent direction.

The assembled unit and details of construction are shown in Figure 1.

Materials and Costs. The cost of this heat exchanger, including the pump and motor but exclusive of the labor involved in construction and erection, is about \$140. A bill of materials is given in Table 1.

This does not include the following auxiliary equipment, which is necessary for making an experimental run:

1. Potentiometer.
2. Stop watch.
3. Platform scale: 50-pound capacity.
4. Weighing bucket: 5-gallon capacity.

Notes on Construction. 1. Thermometers and thermocouples are inserted at various points in the system. The thermometers are inserted in wells constructed of nipples and bushings as shown in Detail A of the drawing. The thermocouples are held in place between flanges, the gaskets of which provide the necessary insulation.

2. The concentric-pipe arrangement is provided by brazing the inner pipe to a bushing, drilled to receive it, and screwed into the tee of the outer pipe. This arrangement is shown in Detail B.

3. The exchanger is completely covered with standard 85 per cent magnesia pipe covering.

4. At *Drexel Institute of Technology*, this apparatus is mounted on a support made of steel channels and attached to the wall.

Operation. Water from reservoir *E* is pumped through the inner pipe of section *D* where it is heated by steam in the annular space. It then enters the annular space of section *C* at *H* and passes successively through the annular spaces of sections *B* and *A*, discharging at *I* to weir box *J*, from which it returns to the reservoir.

* Reported by H. T. Ward.

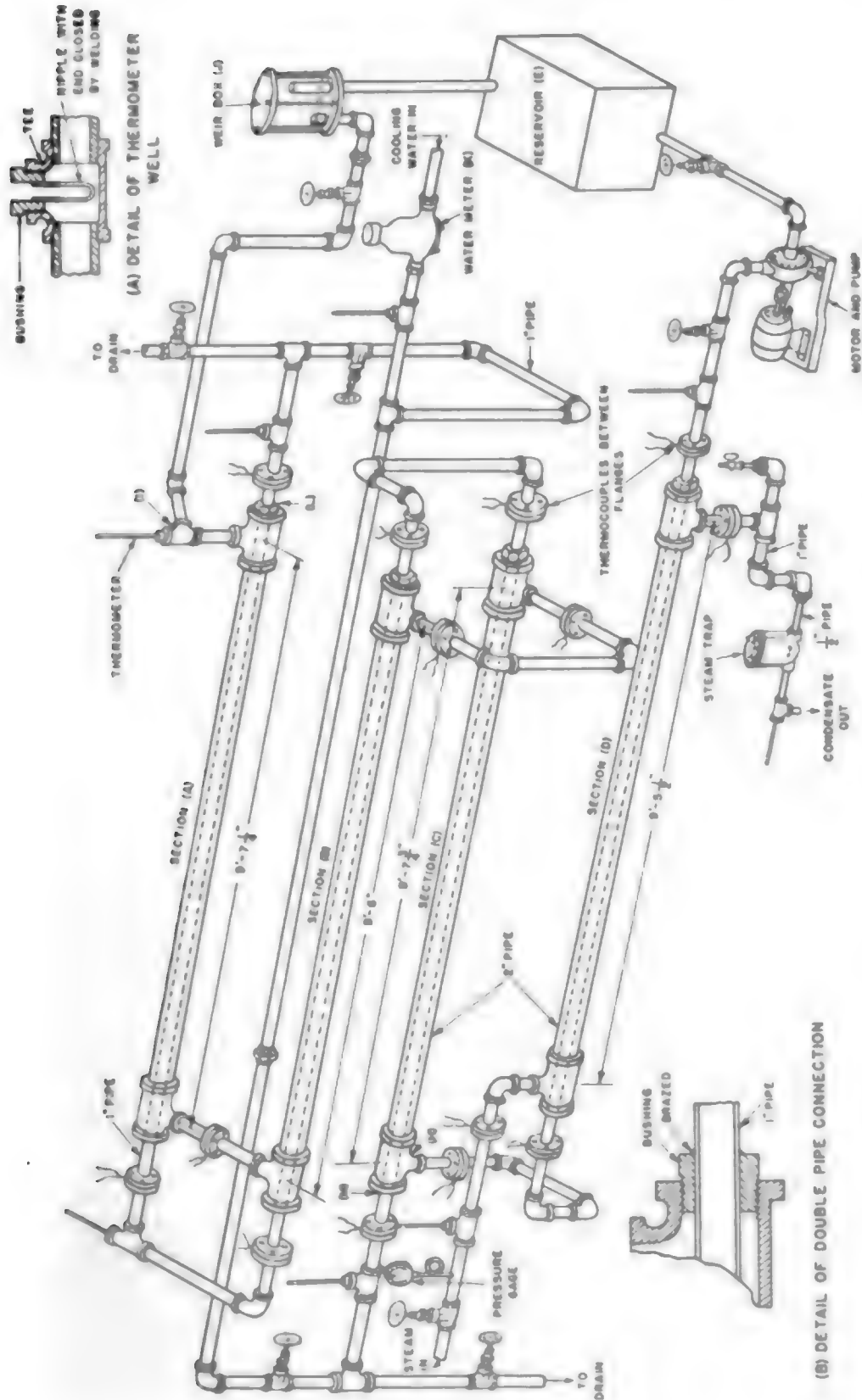


Figure 1. Concentric-Pipe Heat Exchanger—Drexel Institute of Technology

For countercurrent operation, cooling water is passed through water meter *K*, enters the inner pipe of section *A* at *L*, and passes successively

TABLE 1
Bill of Materials for Concentric-Pipe Heat Exchanger
Drexel Institute of Technology

Quantity	Description	Size	Material
38 ft.	Std. pipe	2 in.	Black wrought steel
60 ft.	" "	1 in.	" " "
14	Std. nipple	1 in. x 6 in. long	" " "
31	" "	1 in. x 4 in. "	" " "
7	" "	1 in. x 2 in. "	" " "
2	" "	$\frac{1}{2}$ in. x 4 in. "	" " "
9	" "	$\frac{1}{2}$ in., short	" " "
8	Std. tee	2 in. x 2 in. x 1 in.	Black malleable iron
10	" "	1 in. x 1 in. x 1 in.	" " "
1	" "	$\frac{1}{2}$ in.	" " "
19	Std. 90° elbow	1 in.	" " "
1	" 45° "	1 in.	" " "
2	Ground-joint union	1 in.	" " "
8	Bushing	2 in. x 1 in.	" " "
8	"	1 in. x $\frac{1}{2}$ in.	" " "
1	"	1 in. x $\frac{1}{4}$ in.	" " "
1	"	$\frac{1}{2}$ in. x $\frac{1}{4}$ in.	" " "
14	Std. flange union, gasket type	1 in.	Cast iron
8	Std. gate valve	1 in.	Brass
1	Pet cock	$\frac{1}{2}$ in.	"
28	Gasket	1 in. x $\frac{1}{16}$ in.	Cranite
40 ft.	Pipe insulation	2 in.	85% Magnesia
10 ft.	" "	1 in.	" "
1	Steam trap	$\frac{1}{2}$ in.	Cast iron
1	Pressure gage	0 to 100 lb.; $3\frac{1}{2}$ in. face	Brass
1	Water meter	$\frac{1}{2}$ in.	Bronze
1	Centrifugal pump and motor assembly	1 in.; 13 g.p.m.; $\frac{1}{2}$ H.P.	Bronze-fitted
1	Reservoir	100-gal. capacity	Steel
1	Weir box	1-ft. ³ /min. capacity	Glass and Brass
8	Thermometer	0 to 260°F.	Glass-mercury
200 ft.	Thermocouple wire	24 gage	Copper-constantan

through the inner pipes of sections *B* and *C*, discharging at *M* to the drain.

For parallel flow, the cooling water enters at *M* and discharges at *L*.

TYPICAL EXPERIMENT

Object. To determine the overall heat-transfer coefficient in a double-pipe heat exchanger for: (1) steam to water, (2) water to water in parallel flow, and (3) water to water in countercurrent flow.

Procedure. 1. Determine the method of operating the system.

TABLE 2
Flow Data of Heat-Transfer Experiment

Run No.	Weir reading, ft.	Flow of circulating water, ft. ³ /min. (from weir plot*)	Flow of cooling water, ft. ³ /run. (from meter readings)	Weight of condensate, lb.	Time of run, min.
Parallel flow					
1	0.66	0.88	35.2	25.25	15
2	0.54	0.49	32.2	30.00	20
Countercurrent flow					
3	0.55	0.53	18.5	25.25	15
4	0.67	0.91	58.0	18.00	20

* Provided by instructor.

TABLE 3
Temperature Data of Heat-Transfer Experiment

Symbol	Item	Parallel flow		Countercurrent flow	
		Run No. 1	Run No. 2	Run No. 3	Run No. 4
		Temperature °F.			
t_1	Steam to heating section.....	212.0	214.8	220.2	212.0
t_2	Condensate.....	179.3	193.1	196.2	114.3
t_3	Circulating water to heating section.....	93.3	96.0	97.8	73.5
t_4	Circulating water from heating section.....	124.0	140.4	142.9	91.5
t_5	Circulating water from liquid-liquid section.....	93.7	96.6	98.5	73.9
t_6	Inlet cooling water.....	40.0	40.0	40.3	40.0
t_7	Outlet cooling water.....	51.7	54.6	59.4	46.0

2. Make the measurements necessary for calculating all of the heating surface areas.

3. Adjust the necessary valves to give the proper rates of flow of the steam and the two water streams.

4. After equilibrium conditions have been established, make the measurements of rates of flow and temperature that are necessary in order to calculate the overall coefficient of heat transfer.

5. Make at least three runs each for both parallel and countercurrent flow.

TABLE 4
Experimental and Calculated Results of Heat-Transfer Experiment

Symbol	Item	Steam-heated section (area = 2.593 ft. ² = A ₁)				Liquid-liquid section (area = 7.89 ft. ² = A ₂)			
		Parallel		Countercurrent		Parallel		Countercurrent	
		Run No. 1	Run No. 2	Run No. 3	Run No. 4	Run No. 1	Run No. 2	Run No. 3	Run No. 4
Experimental results									
Δt ₁	Arith.-mean diff., °F.	103.4	96.6	96.0	129.0	63.0	71.2	70.9	39.7
Δt ₂	Log.-mean diff., °F.	104.5	95.0	97.5	128.0	60.6	66.8	69.6	39.4
q	Heat transferred, B.t.u./hr.	101,300	80,500	89,000	57,600	103,000	88,000	87,000	60,000
U _o	Overall coeff.,* (B.t.u./ (hr. × ft. ² × °F.))	373	326	352	174	215	168	160	193
Calculated from film-coefficient equations									
A _s	Steam-film coeff., B.t.u./(hr. × ft. ² × °F.)	1,092	1,205	1,080	1,910	—	—	—	—
A ₁	Inner-film coeff., B.t.u./(hr. × ft. ² × °F.)	659	412	462	591	988	741	609	1,150
A ₂	Outer-film coeff., B.t.u./(hr. × ft. ² × °F.)	—	—	—	—	313	209	727	265
U _c	Overall coeff., B.t.u./ (hr. × ft. ² × °F.)	393	300	352	415	254	178	181	233

* Based on log-mean Δt and inside surface of pipe.

TABLE 5
*Summary of Experimental and Calculated Overall Heat-Transfer Coefficients
 U in B.t.u./(hr. \times ft.² \times °F.)*

Run. No.	Steam-heated section			Liquid-liquid section		
	Experimental U_o	Calculated U_c	Per cent difference*	Experimental U_o	Calculated U_c	Per cent difference*
1	373	393	5.4	215	254	18.2
2	326	300	-8.0	168	178	6.0
3	352	352	0	160	181	13.1
4	174	415	**	193	233	20.8

* Per cent difference is based on the experimental value.

** The results for Run No. 4 were greatly in error because data were taken before equilibrium conditions were attained.

6. Calculate:

- Arithmetic-mean temperature difference.
- Logarithmic-mean temperature difference.

- c. Overall heat-transfer coefficients, for each condition, from the experimental data.
- d. Overall heat-transfer coefficients, for each condition, using film coefficient equations. Compare these with the experimental results.

Typical Data. Experimental data and calculated results for four runs are given in Tables 2 to 4, and the experimentally determined coefficients are compared with those calculated from film coefficient equations in Table 5.

CALCULATION FOR RUN NO. 1 FOR PARALLEL FLOW IN LIQUID-LIQUID SECTION

Overall heat-transfer coefficient from experimental data. This is calculated by means of the equation¹

$$\begin{aligned}
 U_o &= \frac{q}{A_2 \times \Delta t_2} \\
 &= \frac{103,000}{7.89 \times 60.6} = 215 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}).
 \end{aligned}
 \tag{1}$$

where

$$\begin{aligned}
 103,000 &= q = \text{heat transferred, B.t.u./hr.} \\
 &= (t_7 - t_6) \times c_p \times (\text{cu. ft. water/run}) \\
 &\quad \times \rho \times \frac{60}{\text{time of run, min.}}
 \end{aligned}$$

$$= (51.7 - 40.0) \times 1.0 \times 35.2 \times 62.4 \times 60/15$$

$$7.89 = A_2 = \text{area of the liquid-liquid section, ft.}^2 \text{ (from Table 4).}$$

$$60.6 = \Delta t_2 = \text{logarithmic-mean temperature difference, } ^\circ\text{F.}$$

$$\begin{aligned}
 &= \frac{(t_4 - t_6) - (t_6 - t_7)}{2.303 \log_{10} \frac{(t_4 - t_6)}{(t_6 - t_7)}} = \frac{(124.0 - 40.0) - (93.7 - 51.7)}{2.303 \log_{10} \frac{(124.0 - 40.0)}{(93.7 - 51.7)}}
 \end{aligned}$$

Overall heat-transfer coefficient from film coefficient equations. This coefficient is calculated by means of the equation²

$$\begin{aligned}
 U_o &= \frac{1}{\frac{1}{h_1} + \frac{D_1 L}{D_o k} + \frac{D_1}{D_2 h_2}} \\
 &= \frac{1}{\frac{1}{988} + \frac{0.0874 \times 0.0111}{0.0985 \times 26} + \frac{0.0874}{0.1095 \times 313}} \\
 &= 254 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}).
 \end{aligned}
 \tag{2}$$

where

$$D_1 = \text{inside diameter of the pipe, ft.} = \frac{1.049}{12} = 0.0874$$

$$D_2 = \text{outside diameter of the pipe, ft.} = \frac{1.315}{12} = 0.1095$$

$$D_{av} = \text{average diameter of the pipe, ft.} = \frac{1}{2}(0.0874 + 0.1095) = 0.0985$$

$$L = \text{wall thickness of the pipe, ft.} = \frac{1}{2}(0.1095 - 0.0874) = 0.0111$$

$$k = \text{thermal conductivity of the pipe, B.t.u.-ft.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}) = 26$$

$$h_1 = \text{inner-film coefficient, B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}) = 988$$

$$h_2 = \text{outer-film coefficient, B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}) = 313$$

The inner- and outer-film coefficients, h_1 and h_2 , are calculated by means of the Dittus-Boelter equation³

$$h = 0.0225 \left(\frac{k}{D} \right) \left(\frac{DV\rho}{\mu} \right)^{0.8} \left(\frac{c_p \mu}{k} \right)^n \quad (3)$$

where

k = thermal conductivity of the fluid at its average temperature, B.t.u.-ft.}/(hr. / ft.² × °F.).

D = diameter, ft.

V = velocity, ft./hr.

ρ = density of the fluid at its average temperature, lb./ft.³

μ = viscosity of the fluid at its average temperature, lb.}/(hr. × ft.).

c_p = heat capacity of the fluid, B.t.u.}/(lb. × °F.).

n = 0.4 when the liquid is being heated, and 0.3 when it is being cooled.

1. Inner film

$$\begin{aligned} h_1 &= 0.0225 \times \frac{0.330}{0.0874} \\ &\quad \times \left(\frac{0.0874 \times 23,400 \times 62.42}{3.388} \right)^{0.8} \left(\frac{1.0 \times 3.388}{0.330} \right)^{0.4} \\ &= 988 \end{aligned}$$

where

0.330 = k = thermal conductivity of water at the average temperature of 46°F., B.t.u.-ft.}/(hr. × ft.² × °F.).

0.0874 = D = inside diameter of pipe, ft.

23,400 = V = velocity of the water, ft./hr.

$$= (\text{ft.}^3/\text{run}) \times \frac{60}{(\text{min.}/\text{run})} \times \frac{144}{\text{cross-sectional area of the pipe, in.}^2}$$

$$= 35.2 \times \frac{60}{15} \times \frac{144}{0.864}$$

62.42 = ρ = density of water at 46°F., lb./ft.³

3.388 = μ = viscosity of water at 46°F., lb./(ft. \times hr.)
 = viscosity in centipoises $\times 2.42$
 = 1.400 $\times 2.42$

1.0 = c_p = heat capacity of water, B.t.u./(lb. \times °F.).

2. Outer film

$$h_2 = 0.0225 \times \frac{0.366}{0.0627} \left(\frac{0.0627 \times 3,805 \times 61.87}{1.505} \right)^{0.8} \left(\frac{1.0 \times 1.505}{0.366} \right)^{0.3}$$

$$= 313$$

where

0.366 = k = thermal conductivity of water at the average temperature of 109°F., B.t.u.-ft./(hr. \times ft.² \times °F.).

0.0627 = D = equivalent diameter⁴, ft.

$$= 4 \times \frac{\text{cross section of the annulus}}{\text{wetted perimeter}}$$

$$= 4 \times (\text{I. D. of 2 inch pipe} - \text{O. D. of 1 inch pipe})$$

$$= 4 \times \frac{(2.067 - 1.315)}{12}$$

3,805 = V = velocity of the water, ft./hr.

$$= (\text{ft.}^3/\text{min.}) \times 60 \times \frac{144}{\text{cross-sectional area of annulus, in.}^2}$$

$$= 0.88 \times 60 \times \frac{144}{(3.355 - 1.358)}$$

61.87 = ρ = density of water at 109°F., lb./ft.³

1.505 = μ = viscosity of water at 109°F., lb./(ft. \times hr.)
 = 0.622 $\times 2.42$

NOMENCLATURE

A = area, ft.²

c_p = heat capacity, B.t.u./(lb. \times °F.).

D = diameter, ft.

h = film coefficient, B.t.u./($\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}$).

k = thermal conductivity, B.t.u.-ft./($\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}$).

n = exponent in the Prandtl number.

q = heat transferred, B.t.u./hr.

t = temperature, $^\circ\text{F.}$

Δt = temperature difference, $^\circ\text{F.}$

U = overall heat-transfer coefficient, B.t.u./($\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}$).

V = velocity, ft./hr.

μ = viscosity, lb./($\text{ft.} \times \text{hr.}$).

ρ = density, lb./ft.³

Subscript

1 = inside.

2 = outside.

av = average.

c = calculated.

e = experimental.

REFERENCES

1. BADGER AND McCABE, "Elements of Chemical Engineering," McGraw-Hill Book Company, New York, 1936, 2nd ed., p. 129.
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3. Ibid., p. 134.
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A CONCENTRIC-PIPE HEAT EXCHANGER

Constructed by

The Department of Chemical Engineering and Chemistry
Missouri School of Mines and Metallurgy*

Description. This apparatus (Figure 1), which is used principally for comparing calculated mean temperature differences with the actual mean, consists, essentially, of six sections of concentric pipes welded in series.** Hot water is passed through the inner pipe and cold water through the annular opening between the pipes. Valves are provided for reversing the direction of the cold stream to secure either countercurrent or parallel flow; and orifice meters are used to measure the rate of flow of each stream. Each section is provided with thermometer wells at the entrance and exit so that corresponding temperature readings of both streams can be taken at seven equally spaced points over the entire paths of the fluids.

Materials and Cost. This apparatus is constructed almost entirely of standard pipe and fittings. The materials, exclusive of the thermometers, cost a total of only about \$22; these materials are listed in Table 1.

Notes on Construction. 1. Since the streams are separated at the ends of each four-foot section to permit the insertion of thermometer wells, these points should be well insulated.

2. The exchanger is mounted in a horizontal position on a frame made of standard pipe.

Comments on Design. 1. Although the external thermometer wells and the concentric-pipe arrangement are of welded construction, the entire apparatus may be made from threaded pipe and standard fittings.

2. The orifice meters may be eliminated and flow-rate determinations may be made by weighing the fluids.

TYPICAL EXPERIMENT

Object. To determine the actual mean temperature difference between hot and cold water in both countercurrent and parallel flow, and to compare the experimental values with the calculated arithmetic and logarithmic-mean differences.

* Reported by F. H. Conrad.

** Based on a design by W. L. Beuschlein, Department of Chemical Engineering, University of Washington, Seattle Washington.

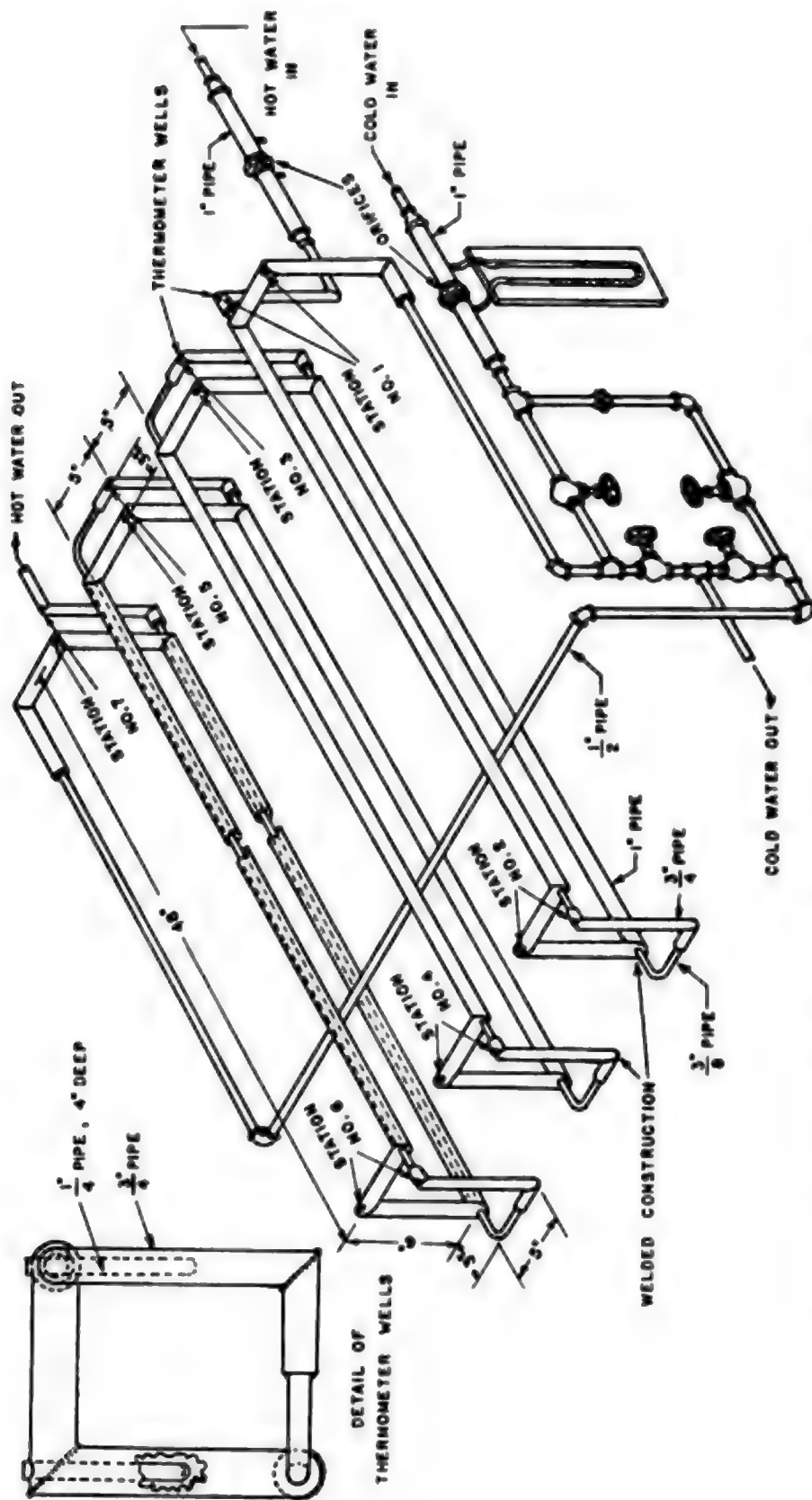


Figure 1. Concentric-Pipe Heat Exchanger—Missouri School of Mines and Metallurgy

Procedure. 1. Set the valves for countercurrent flow and adjust them so that the two streams have approximately the same rate of flow.

2. Read the thermometers occasionally until thermal equilibrium is attained; and then obtain all the data necessary for plotting the heat-transfer driving force as a function of exchanger length.

3. Adjust the flow rates so that one liquid stream has approximately twice the throughput of the other and repeat the experiment.

4. Make two similar runs with parallel flow.

TABLE 1
Bill of Materials for Concentric-Pipe Heat Exchanger
Missouri School of Mines and Metallurgy

Quantity	Description	Size	Material
36 ft.	Std. pipe	1 in.	Black wrought steel
10 ft.	" "	$\frac{1}{2}$ in.	" " "
22 ft.	" "	$\frac{1}{2}$ in.	" " "
29 ft.	" "	$\frac{1}{2}$ in.	" " "
5 ft.	" "	$\frac{1}{2}$ in.	" " "
1	Std. nipple	$\frac{1}{2}$ in. x $4\frac{1}{2}$ in. long	" " "
7	" "	$\frac{1}{2}$ in. x 3 in. "	" " "
8	" "	$\frac{1}{2}$ in., short	" " "
2	" "	$\frac{1}{2}$ in. x $4\frac{1}{2}$ in. long	" " "
1	" "	$\frac{1}{2}$ in., short	" " "
4	Std. tee	$\frac{1}{2}$ in.	Black malleable iron
17	Std. 90° elbow	$\frac{1}{2}$ in.	" " "
1	" 45° "	$\frac{1}{2}$ in.	" " "
1	Std. 45° street elbow	$\frac{1}{2}$ in.	" " "
3	Ground-joint union	$\frac{1}{2}$ in.	" " "
2	" " "	$\frac{1}{2}$ in.	" " "
2	Bushing	1 in. x $\frac{1}{2}$ in.	Cast iron
1	Std. coupling	1 in.	Black malleable iron
4	Low-pressure globe valve	$\frac{1}{2}$ in.	Brass
2	Meriam manometer	15 in. long	Steel and glass
14	Thermometer	0 to 220° F.	Glass-mercury

Report and Discussion. 1. From the experimental data for each run, plot the temperature difference at each station as a function of exchanger length, and calculate the true mean temperature difference by graphical integration.

2. From the terminal conditions for each run, calculate the arithmetic- and logarithmic-mean temperature differences.

3. Compare these values with the true mean temperature difference

in each case, and calculate the percentage error introduced through the use of the arithmetic- or logarithmic-mean. Show when the arithmetic- or logarithmic-mean may be used. Discuss the use of each with regard to these particular data.

TABLE 2
Experimental Heat Transfer Data for Parallel and Countercurrent Flow

Run No.	1		2		3		4	
Flow	Parallel		Countercurrent		Countercurrent		Parallel	
Stream	Hot	Cold	Hot	Cold	Hot	Cold	Hot	Cold
Manometer, in. Hg	1.0	1.0	1.0	1.0	0.85	1.95	0.95	2.0
Flow, rate, gal./min.	2.05	2.20	2.05	2.20	1.90	3.10	2.00	3.17
Station*	(Temperature, °F.)							
1	195	56	188	125	193	112	189	56
2	171	74	175	103	174	101	177	71
3	157	89	164	101	159	85	151	84
4	144	101	142	88	144	80	138	90
5	139	107	141	77	131	70	130	98
6	135	114	131	67	120	63	124	102
7	131	117	121	56	110	56	120	101

* Distance between thermometer stations = 4 ft.; total length of exchanger = 24 ft.

TABLE 3
Calculated Results of Heat Transfer Experiments

Run No.	$\int_0^M \Delta M L$	True mean Δt , °F.	Δt , Station 1, °F.	Δt , Station 7, °F.	Mean Δt , °F.		Per cent error	
					Log.	Arith.	Log. mean	Arith. mean
1	1360	56.67	139	14	56.5	76.5	-0.3	35.0
2	1543	64.33	63	65	63.5	64.0	-1.3	-0.5
3	1600	66.67	81	54	66.6	67.5	-0.1	1.3
4	1375	57.33	133	16	55.2	74.5	-3.8	30.0

4. Mathematically, upon what conditions is the logarithmic-mean based? Derive the logarithmic-mean equation.

5. Discuss briefly the relative merits of countercurrent and parallel flow of fluids for the transfer of heat.

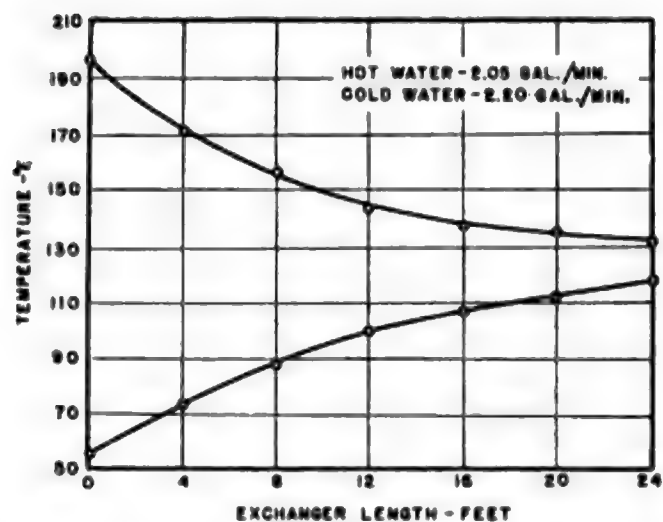


Figure 2. Temperature Readings for Run No. 1, Parallel Flow

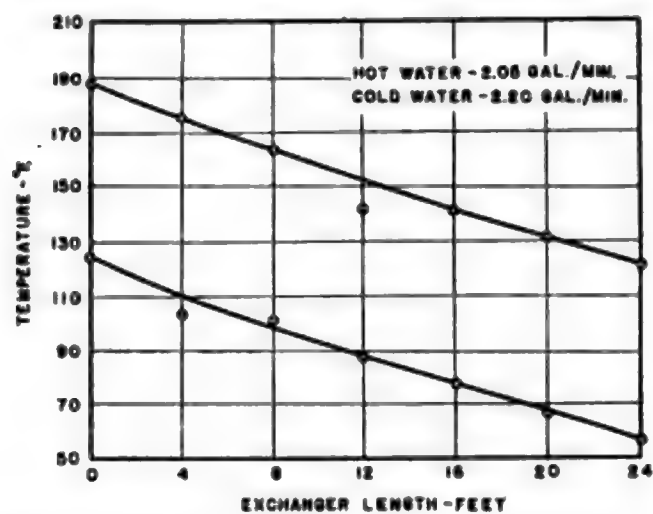


Figure 3. Temperature Readings for Run No. 2, Countercurrent Flow

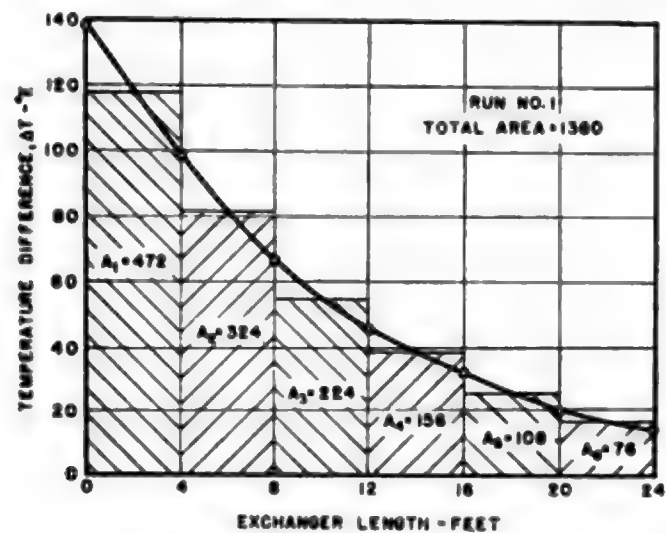


Figure 4. Temperature Differences and Results of Graphical Integration

Experimental Data and Calculated Results. The experimental data for two parallel and two countercurrent runs are given in Table 2, and the calculated results are given in Table 3.

The temperature readings for Runs 1 and 2 are plotted against exchanger length in Figures 2 and 3, respectively, and the temperature differences for Run No. 1 are plotted in Figure 4.

Calculations for Run No. 1.

1. *True mean temperature difference.*

The temperature difference for each station was read from Figure 2, and the values were plotted against exchanger length, as shown in Figure 4. This curve was then integrated graphically; the results are:

$$\int_0^{24} \Delta t dL = \text{area under the curve} = 1360$$

$$\begin{aligned} \text{therefore, the true mean } \Delta t &= \frac{1360}{\text{exchanger length}} \\ &= \frac{1360}{24} = 56.67^\circ\text{F.} \end{aligned}$$

2. *Log-mean temperature difference*

$$\begin{aligned} &= \frac{\Delta t \text{ for station 1} - \Delta t \text{ for station 7}}{2.303 \log_{10} \left(\frac{\Delta t \text{ for station 1}}{\Delta t \text{ for station 7}} \right)} \\ &= \frac{139-14}{2.303 \log_{10} \left(\frac{139}{14} \right)} = 56.5^\circ\text{F.} \end{aligned}$$

3. *Arithmetic-mean temperature difference*

$$\begin{aligned} &= \frac{1}{2}(\Delta t \text{ for station 1} + \Delta t \text{ for station 7}) \\ &= \frac{1}{2}(139 + 14) = 76.5^\circ\text{F.} \end{aligned}$$

4. *Per cent deviation from true mean Δt .*

$$\begin{aligned} \text{a. Log-mean error, \%} &= \frac{56.5 - 56.67}{56.67} \times 100 \\ &= \frac{-0.17}{56.67} \times 100 = -0.3 \end{aligned}$$

$$\begin{aligned} \text{b. Arithmetic-mean error, \%} &= \frac{76.5 - 56.67}{56.67} \times 100 \\ &= \frac{19.83}{56.67} \times 100 = 35.0 \end{aligned}$$

A SINGLE-PASS DOUBLE-PIPE HEAT EXCHANGER

Designed and Constructed by
The Department of Chemical Engineering
University of Arkansas*

Description. This concentric-pipe heat exchanger is made of 2-inch and 1½-inch standard pipe and fittings, and is arranged in such a manner that liquid flowing through the inner pipe can be heated by either steam or hot water flowing through the annular space in countercurrent direction. The assembly of this apparatus is shown in Figure 1.

Materials and Cost. This heat exchanger was constructed at a cost of approximately \$55, exclusive of labor; the materials are listed in Table 1.

Notes on Construction. *Thermometer Wells.* The details of construction of thermometer wells T_1 to T_4 are shown in Figure 1. Each well consists of a short nipple; the lower end of the nipple is closed by welding, and the upper end fits into a bushing that has been threaded on the inside from the bottom. The thermometer bulb is held rigidly by a cork in the mouth of the well, and its bulb is surrounded by mercury to insure good thermal contact.

Orifice Meters. The orifice meters O_1 and O_2 are made of brass plates, ⅛-inch thick, held between gaskets in standard 1½-inch cast-iron flange unions. The ratio of orifice diameter to pipe diameter is 0.421 for both meters.

Water-Steam Mixer. To minimize the water-hammer effect, a perforated plug is inserted in the steam line at A, the point where the steam and water mix.

Comments on Design. 1. This unit is constructed to operate either as a steam heater or as a countercurrent heat exchanger. With slight modifications it can be made to operate in parallel flow.

2. Loop P is provided to help keep the apparatus free of air, which otherwise might interfere with proper manometer operation.

3. Orifice meters O_1 and O_2 are installed in lines carrying heated water, the temperature of which varies with the water-flow rate and the pressure of the steam; if the orifices are calibrated at one temperature only, the calibration might be expected to be somewhat in error at other temperatures. However, with a ratio of orifice to pipe diameters

* Reported by the late R. W. Higbie.

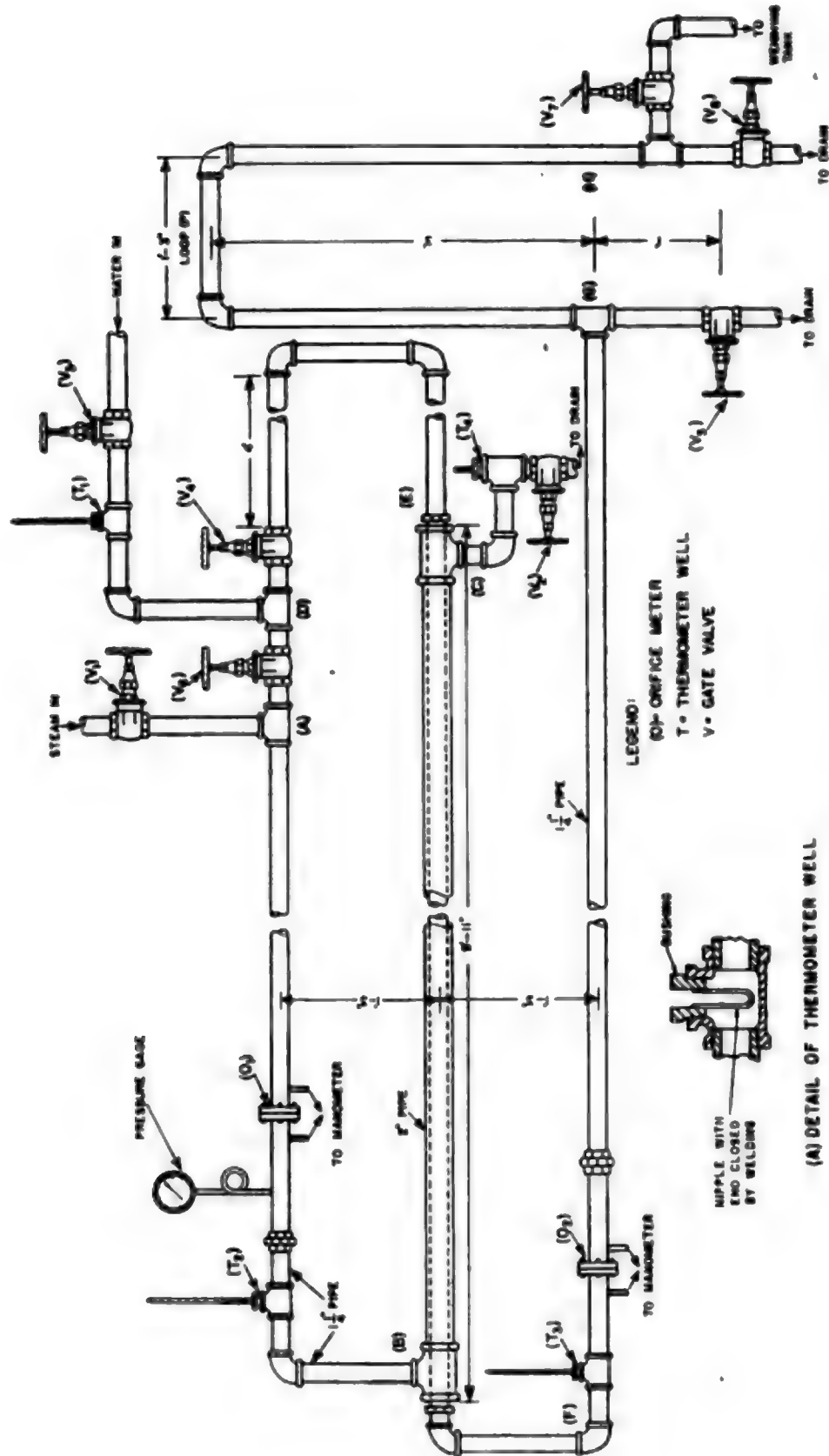


Figure 1. Single-Pass, Double-Pipe Heat Exchanger—University of Arkansas

of 0.421 and the range of Reynolds' number encountered in this apparatus, this effect is negligible.¹

Operation. *Using Steam in the Outer Jacket.* Steam, introduced through valve V_1 , enters the annular space through line $A-B$. The condensate leaves the annular space at C and passes to the drain through a line containing thermometer well T_4 and valve V_2 , which is

TABLE 1
Bill of Materials for Double-Pipe Heat Exchanger
University of Arkansas

Quantity	Description	Size	Material
8 ft., 8 in.	Std. pipe	2 in.	Black wrought steel
51 ft.	" "	1½ in.	" " "
6	Std. nipple	1½ in. x 12 in. long	" " "
14	" "	1½ in. x 6 in. "	" " "
1	" "	1½ in., close	" " "
4	" "	½ in. x 2 in. long	" " "
8	Std. tee	1½ in.	Black malleable iron
2	Double-pipe tee	2 in. x 1½ in. x 1½ in.	Ferroteel
10	Std. 90° elbow	1½ in.	Black malleable iron
2	Ground-joint union	1½ in.	" " "
4	Bushing	1½ in. x ½ in.	" " "
2	Std. flange union, gasket type	1½ in.	Cast iron
8	Std. gate valve	1½ in.	Brass
1	Pressure gage	0 to 35 lb.; 3½-in. face	"
2	Manometer	20 in. long; 8-mm. tubing	Glass-mercury
4	Thermometer	0 to 220°F.	"

opened slightly to allow a small amount of steam to pass out with the condensate.

Water, introduced through valve V_3 , enters the inner pipe through line $D-E$ and passes either to the drain through valve V_6 or to a weighing tank through valve V_7 .

Valve V_4 is used only to drain the apparatus; it is kept closed during the run.

Using Warm Water in Outer Jacket. The flow of water is divided at D , and the rates of the two streams are regulated by adjusting valves V_4 and V_6 . The water in the inner pipe follows the path pre-

viously described; the water in the annular space is first heated by being mixed with live steam at A.

TYPICAL EXPERIMENT

Object. To determine the overall coefficient of heat transfer from condensing steam to water in forced convection.

Procedure. 1. Fill the water line and purge the manometer leads of any air bubbles, pass water through the inner pipe at the desired rate of flow, introduce steam into the annulus at the desired pressure, and crack the condensate valve slightly to allow a small amount of steam to pass out with the condensate.

2. After conditions of flow and temperature have become constant, make a run for at least 20 minutes, recording the following data about every 2 minutes:

- a. Time.
- b. Orifice manometer readings.
- c. Temperatures of the inlet water and exit water.
- d. Steam pressure.

3. Repeat the experiment at different steam temperatures and different water flow rates.

4. Calculate the overall coefficient of heat transfer from the data for each run.

Typical Data and Calculations.

The experimental data for a typical run are given in Table 2.

The calculation of the overall heat-transfer coefficient was made as follows:

Item 1: *Temperature of steam at 1.1 lb./in.² gage (15.8 lb./in.² abs.)*
 = 215.7 °F. (from steam table).

Item 2: *Flow rate of the water*
 = Orifice constant $\sqrt{\text{manometer reading}}$
 = $9.3 \sqrt{202} = 132 \text{ lb./min.}$

Item 3: *Heat absorbed per lb. of water*
 = (exit water temp. - inlet water temp.) \times heat capacity of water
 = $(95.8 - 48.2) \times 1.0 = 47.6 \text{ B.t.u.}$

Item 4: *Heat absorbed per hour, q*
 = Item 2 \times 60 \times Item 3
 = $132 \times 60 \times 47.6 = 377,000 \text{ B.t.u./hr.}$

TABLE 2

*Experimental Data for Heat Transfer Between Steam and Water*Inside heating surface, ft.² = 3.52*Orifice constant, lb./min. $\times \frac{1}{\sqrt{\text{mm. Hg}}} = 9.3^*$

Time, P.M.	Manometer reading, mm. Hg	Water temperature		Steam pressure, lb./in. ² gage.
		Inlet, °F.	Exit °F.	
3:18	395-595	48.3	95.5	1.0
3:20	400-590	48.2	96.2	1.0
3:22	395-595	48.3	95.5	1.0
3:24	395-595	48.3	96.0	1.0
3:26	390-600	48.2	95.6	1.0
3:28	390-600	48.2	95.7	1.1
3:30	400-590	48.2	95.8	1.1
3:32	390-600	48.2	95.8	1.1
3:34	395-595	48.2	96.0	1.3
3:36	390-600	48.2	96.0	1.4
Average.....	202.0 (diff.)	48.2	95.8	1.1

* These data were furnished by the instructor.

Item 5: *Log-mean temperature difference, Δt_m .*

$$\Delta t_m = \frac{(215.7 - 48.2) - (215.7 - 95.8)}{2.303 \log_{10} \frac{(215.7 - 48.2)}{(215.7 - 95.8)}} = 142.6^\circ\text{F.}$$

Item 6: *Overall coefficient of heat transfer, U .*

$$U = \frac{\text{Item 4}}{\text{Heating surface} \times \text{Item 5}} = \frac{377,000}{3.52 \times 142.6} \\ = 710 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}).$$

REFERENCE

1. PERRY, "Chemical Engineers' Handbook," McGraw-Hill Book Company, New York, 1941, 2nd ed., p. 852.

A TUBULAR CONDENSER

Designed and Constructed by

The Department of Chemical Engineering
Newark College of Engineering*

Description. This small condenser has a shell made from 2½-inch standard steel pipe and fittings, and a condensing surface constructed of a number of parallel ¾-inch copper tubes soldered into a fixed tube sheet at one end and a floating head at the other end; its construction is shown in Figure 1.

Materials and Cost. This condenser can be constructed at a material cost of approximately \$9. The required materials are listed in Table 1.

Notes on Construction. 1. The shell is made from a 32-inch length of 2½-inch standard pipe, two caps, a nipple, and a flange union. Four holes are drilled and tapped to ½-inch I.P.S. to provide the necessary inlets and outlets: two of the holes are in the pipe and two are in one of the caps.

2. The tube bundle consists of 26 lengths of ¾-inch I.D. by 1⅛-inch O.D. copper tubing, 29 inches long, soldered into two tube sheets; it is provided with a floating head and a return pass. The floating head is made from a 1½-inch length of 2-inch I.P.S. brass pipe, soldered to ¾-inch brass plates. The return pass is a length of ½-inch type M hard copper tubing.

3. Two brass solder-joint couplings (½-inch copper to ½-inch outside I.P.S.) are used to attach the ½-inch copper tubes to the iron cap. It is necessary to drill out one of the couplings to permit the return pass to extend completely through it.

4. It is important to tin and wipe all surfaces to be soldered before the various parts are assembled. In assembling the tube bundle, it is desirable to set all tubes in place in both tube sheets before soldering to either sheet. The tube sheets can then be heated with a torch and more solder applied. The floating head can be soldered together after the tubes are in place.

5. Before the condenser is assembled, the tube bundle should be tested for leaks by filling it with water.

6. After the condenser is assembled and the flanges are tightened, the return pipe is soldered to the solder-joint coupling.

* Reported by A. S. Kohler.

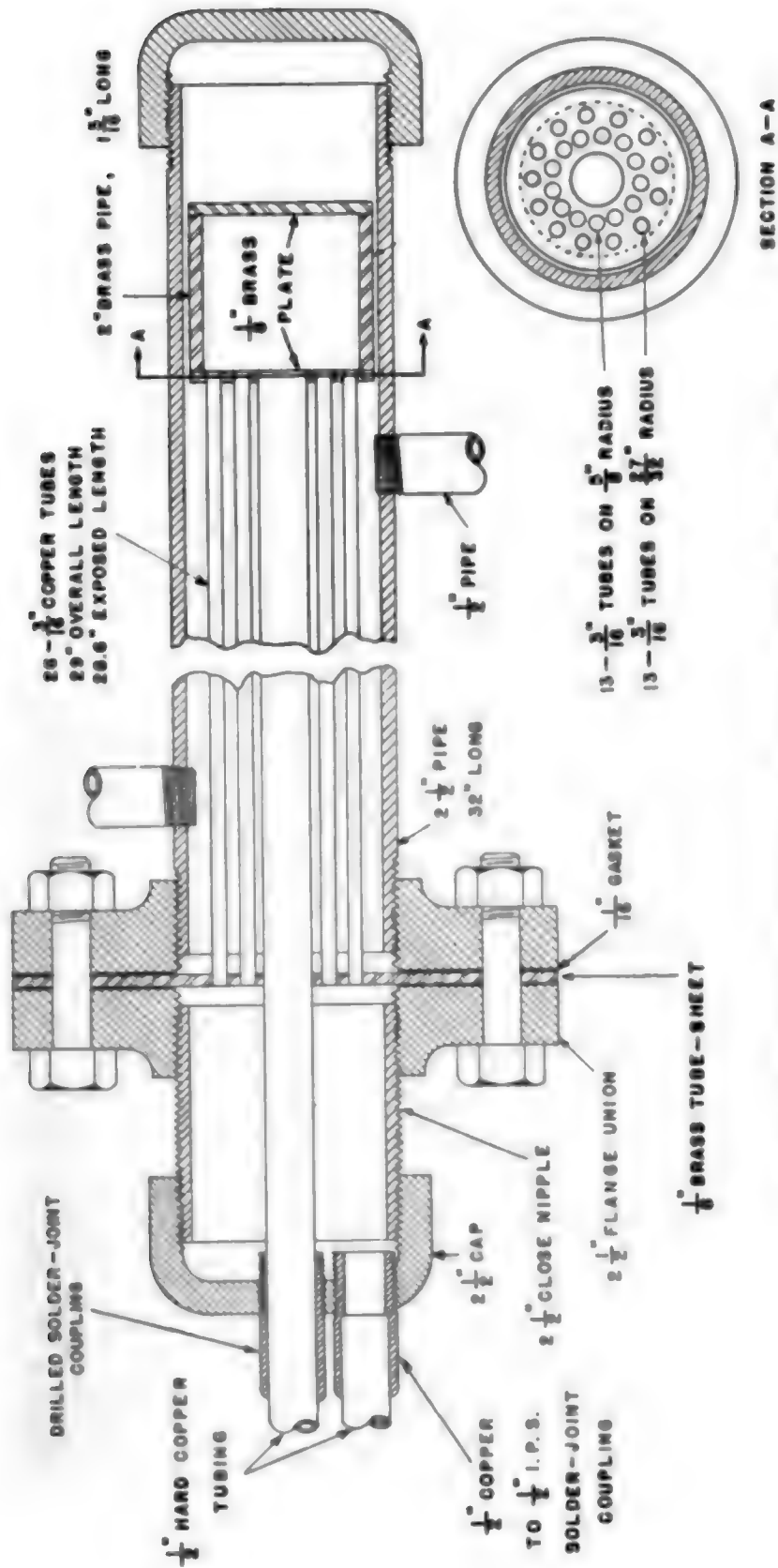


Figure 1. Tubular Condenser—Newark College of Engineering

7. The condenser should be mounted at a slight angle—about 10° to the horizontal—to permit proper drainage.

Comments on Design. A condenser of this design has a number of advantages: (1) it is easy to construct; (2) it is low in cost; (3) the tube bundle can be easily removed for cleaning; and (4) the unit has a very high capacity in proportion to its size, for it is designed for high velocity of water through the tubes and, therefore, the heat-transfer coefficients are higher than usual.

TABLE 1
Bill of Materials for Tubular Condenser
Newark College of Engineering

Quantity	Description	Size	Material
32 in.	Std. pipe	2½ in.	Black wrought steel
1	Std. nipple	2½ in., close	" " "
2	Standard cap	2½ in.	Black malleable iron
1	Std. flange union, gasket type	2½ in.	Cast iron
65 ft.	Tubing	½ in. I.D. x ⅝ in. O.D.	Soft copper
3 ft.	"	½ in.	Hard copper, type M
2	Solder-joint coupling	½ in. copper to ½ in. outside I.P.S.	Brass
2 in.	Pipe	2 in. I.P.S.	"
1	Plate	12 x 12 x ½ in.	"
2	Gasket	7 in. O.D. x ⅝ in. thick	Cranite

TYPICAL EXPERIMENT

Objects. 1. To study the capacity of the condenser as a function of the quantity of cooling water used.

2. To determine the overall heat-transfer coefficients.

Auxiliary Equipment. In addition to the condenser, the following equipment is needed to carry out this experiment:

1. Thermometers: 0 to 220°F ., 2 required.
2. Platform scale: 50-pound capacity.
3. Steam trap: ½-inch size; 250-pound per hour capacity.
4. Weighing bucket: 5-gallon capacity.
5. Water meter: ½-inch size.
6. Stop watch.

Procedure. 1. Run cooling water through the condenser and adjust the rate of flow to the desired value.

2. Introduce steam of known quality at the desired pressure.
3. Read the thermometers occasionally until the system has reached equilibrium.
4. Collect and weigh the condensate over an interval of several minutes, and read the thermometers occasionally during this period.

TABLE 2
Experimental Data of Condenser Tests

Run No.	Water rate, gal./hr.	Water temp., °F.		Steam condensed at 212°F., lb/hr.
		Inlet	Exit	
1	31.2	45	174	40.9
2	147.0	45	156	147.0
3	202.0	45	163	216.0

TABLE 3
Calculated Radiation Losses from Condenser

Run No.	Water rate, lb./hr.	Temp. rise of water, °F.	Heat picked up by water, B.t.u./hr.	Heat given up by condensing steam, B.t.u./hr.	Heat lost by radiation, B.t.u./hr.	Heat lost by radiation, per cent
1	260	129	33,570	39,700	6,130	15.45
2	1227	111	136,300	142,500	6,200	4.35
3	1685	118	198,800	209,500	10,700	5.31

TABLE 4
Calculated Velocities and Overall Heat-Transfer Coefficients

Run No.	Velocity through tubes, ft./sec.	Heat transferred, B.t.u./hr.	Area, ft. ²	Log-mean temp. difference, Δt_m , °F.	Overall heat-transfer coefficient, U , Btu./(hr. \times ft. ² \times °F.)
1	0.523	33,570	3.038	87.2	127
2	2.460	136,300	3.038	101.8	441
3	3.380	198,800	3.038	96.1	679

5. Repeat the experiment at various rates of water flow and at various steam pressures.

6. Calculate the overall heat-transfer coefficient for each condition.

Typical Data from a Report. Experimental data of the rates of flow and temperatures for three runs are given in Table 2. Calculated results for radiation losses, heat transfer coefficients, and velocities through the tubes are given in Tables 3 and 4. The effect of the velocity

of water through the tubes on the overall heat-transfer coefficient is shown in Figure 2.

Calculations for Run No. 1.

1. *Water flow rate, lb./hr.*

$$= (\text{gal./hr.}) \times 231/1728 \times \text{density in lb./ft.}^3$$

$$= 31.2 \times 231/1728 \times 62.4 = 260$$
2. *Temperature rise of the water, °F.*

$$= (\text{exit water temp. °F.}) - (\text{inlet water temp. °F.})$$

$$= 174 - 45 = 129$$

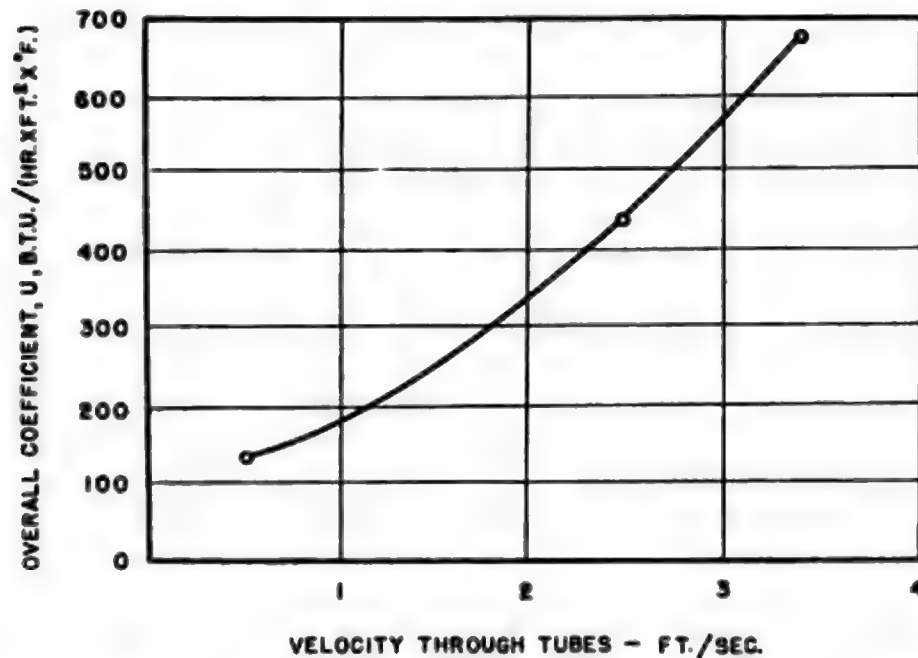


Figure 2. Effect of Water Velocity on Overall Heat-Transfer Coefficient

3. *Heat picked up by the water, B.t.u./hr. = q*

$$= (\text{lb. water/hr.}) \times (\text{temp. rise of water}) \times (\text{heat capacity of water})$$

$$= 260 \times 129 \times 1.0 = 33,570$$
4. *Latent heat of steam at 212°F., B.t.u./lb.*

$$= 970.2 \text{ (from steam tables).}$$
5. *Heat given up by the condensing steam, B.t.u./hr.*

$$= (\text{lb. steam condensed}) \times (\text{latent heat of water at 212°F.})$$

$$= 40.9 \times 970.2 = 39,700$$
6. *Heat lost by radiation, B.t.u./hr.*

$$= (\text{heat given up by condensing steam}) - (\text{heat picked up by water})$$

$$= 39,700 - 33,570 = 6,130$$

7. Heat lost by radiation, per cent

$$= \frac{6,130}{39,700} \times 100 = 15.45$$

8. Log-mean temperature difference, °F. = Δt_m

$$= \frac{(212 - 45) - (212 - 174)}{2.303 \log_{10} \frac{(212 - 45)}{(212 - 174)}} = 87.2$$

9. Area of condensing surface, ft.² = A

= sum of areas of tubes, return pass, floating head, and tube sheets:

- a. Twenty-six copper tubes: 0.125-in. I.D., 0.187-in. O.D., 28.6-in. effective length.

$$d_m = \frac{(0.187 - 0.125)}{2.303 \log_{10} \frac{(0.187)}{0.125}} = 0.155 \text{ in.}$$

$$\text{Area} = \frac{26 \times 28.6 \times \pi \times 0.155}{144} = 2.52 \text{ ft.}^2$$

- b. Return pass: 0.569-in. I.D., 0.625-in. O.D., 28.6-in. long.

$$d_m = \frac{1}{2}(0.625 + 0.569) = 0.597 \text{ in.}$$

$$\text{Area} = \frac{\pi \times 0.597 \times 28.6}{144} = 0.374 \text{ ft.}^2$$

- c. Floating head: 2.067-in. I.D., 2.375-in. O.D., 1.312-in. long.

$$d_m = \frac{1}{2}(2.375 + 2.067) = 2.221 \text{ in.}$$

Area (including end)

$$= \frac{\pi \times 2.221 \times 1.312}{144} + \frac{\pi \times (2.375)^2}{4 \times 144}$$

$$= 0.0636 + 0.0307 = 0.0943 \text{ ft.}^2$$

- d. Tube sheet of the floating head

$$= \left(\frac{\pi \times (2.375)^2}{4 \times 144} \right) - \left(\frac{\pi \times (0.187)^2 \times 26}{4 \times 144} \right) - \left(\frac{\pi \times (0.625)^2}{4 \times 144} \right)$$

$$= 0.0308 - 0.00495 - 0.00213 = 0.0237 \text{ ft.}^2$$

- e. Tube sheet of the fixed head

$$\begin{aligned}
&= \left(\frac{\pi \times (2.469)^2}{4 \times 144} \right) - \left(\frac{\pi \times (0.187)^2 \times 26}{4 \times 144} \right) \\
&\quad - \left(\frac{\pi \times (0.625)^2}{4 \times 144} \right) \\
&= 0.0332 - 0.00495 - 0.00213 = 0.0261 \text{ ft.}^2
\end{aligned}$$

Therefore, the total area of the condensing surface is:

a. Tubes	= 2.52	ft. ²
b. Return pass	= 0.374	"
c. Floating head	= 0.0943	"
d. Tube sheets		
(1) floating head	= 0.0237	"
(2) fixed head	= 0.0261	"
Total	3.0381	ft. ²

10. *Velocity through the tubes, ft./sec.*

$$\begin{aligned}
&= \frac{\text{ft.}^3/\text{sec.}}{\text{cross-sectional area of 26 tubes, ft.}^2} \\
&= \frac{(\text{gal./hr.}) \times (1/3600) \times (231/1728)}{\frac{\pi \times (0.125)^2 \times 26}{4 \times 144}} \\
&= (\text{gal./hr.}) \times (1.68 \times 10^{-3}) = 31.2 \times (1.68 \times 10^{-3}) = 0.523
\end{aligned}$$

11. *Overall heat-transfer coefficient, B.t.u./(hr. \times ft.² \times °F.) = U*

$$= \frac{q}{A \times \Delta t_m} = \frac{33,570}{3.038 \times 87.2} = 127$$

AN APPARATUS FOR TESTING PIPE INSULATION

Designed and Constructed by
The Department of Chemical Engineering
Northeastern University*

Description. This apparatus (Figure 1), which is constructed of standard pipe and fittings, consists of: (1) a steam-heated test section made of a length of 1-inch pipe covered with 3 feet of the insulation through which the heat loss is to be measured, and (2) a smaller length of bare pipe which serves to compensate for loss of heat through any uncovered areas of the pipe carrying the insulation. The steam condensate from each of these sections is passed through jacketed coolers and weighed. Thermocouples, attached to the surface of the pipe and to the surface of the insulation, and thermometers, placed in the surrounding air space, provide the temperature measurements for making the necessary calculations.¹

Materials and Cost. The materials required for the construction of the apparatus are listed in Table 1. The cost of these materials is approximately \$12.

Auxiliary Equipment. The following equipment is needed for experimental work with this apparatus:

1. Balance: 1000-gram capacity.
2. Stop watch.
3. Four glass beakers: 500-ml. capacity.
4. Barometer.
5. Potentiometer: Leeds and Northrup portable, or similar.
6. Thermometers: 0–220°F.

Notes on Construction. 1. To permit proper drainage of the condensate, the apparatus is mounted with a slight downward slope toward each end.

2. The copper-constantan tube-wall thermocouple is attached to the outside surface of the 1-inch pipe by filing a groove and soldering the thermocouple in place; the surface is then filed smooth. The thermocouple for the insulation is wired to the surface.

3. An entrainment separator, made of standard pipe as shown in the drawing, is placed in the steam line to remove as much moisture as possible.

4. To prevent undue air currents, the entire unit is surrounded by an insulated enclosure.

* Reported by C. P. Baker.

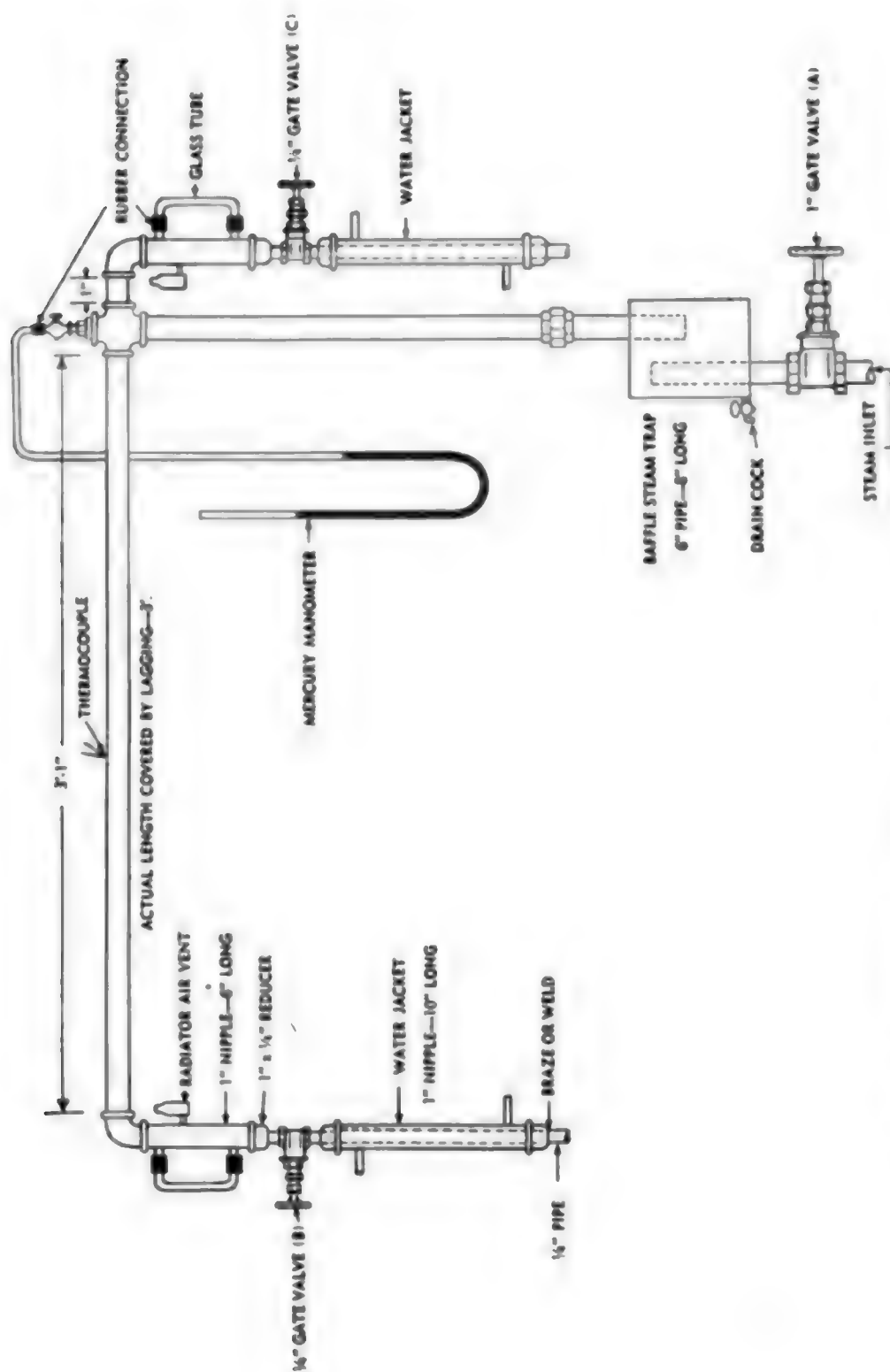


Figure 1. Apparatus for Testing Pipe Insulation—Northern University

5. All parts of the apparatus which carry steam, with the exception of the test section, are covered with 85 per cent magnesia.

Comments on Design. 1. This apparatus can be used to study the heat loss from pipe and insulation into still air, or it can be used to study the effect of air currents on the heat loss; in the latter case, air at known velocities is passed over the surface.

TABLE 1
Bill of Materials for Pipe-Insulation Test Apparatus
Northeastern University

Quantity	Description	Size	Material
8 in.	Std. pipe	6 in.	Black wrought steel
6 ft., 4 in.	" "	1 in.	" " "
2	Std. nipple	1 in. x 10 in. long	" " "
2	" "	1 in. x 6 in. "	" " "
1	" "	1 in. x 1½ in. "	" " "
2	" "	1½ in. x 12 in. "	" " "
2	" "	½ in., close	" " "
6	" "	½ in. x 1½ in. long	" " "
4	" "	½ in. x 1 in. "	" " "
2	Std. 90° elbow	1 in.	Black malleable iron
1	Ground-joint union.	1 in.	" " "
1	Bushing	1 x ½ in.	" " "
2	Std. reducer	1 x ½ in.	" " "
4	Std. cap	1 in.	" " "
1	Std. cross	1 in.	" " "
1	Std. gate valve	1 in.	Brass
2	" " "	½ in.	"
2	Radiator air vent	½ in.	"
1	Pet cock	½ in.	"
1	Drain cock	½ in.	"
1	Manometer	8 mm.	Glass
5 ft.	Pipe insulation	1 in.	85% magnesia
10 ft.	Thermocouple wire	24-gage	Copper-constantan

2. During experiments in which the air is not circulated, the temperature of the surrounding air increases; for very accurate work it may be desirable to control this temperature.

3. The steam leaving the entrainment separator usually has a quality of about 99 per cent. However, for accurate results the quality should be determined; if a throttling calorimeter is not available, it is desirable to superheat the steam slightly before it enters the system.

4. As the quantity of condensate is small, care must be taken to

prevent re-evaporation. This is effectively accomplished by means of the water-jacketed coolers.

Operation. Steam is passed into the system, and when the thermocouples and thermometers indicate that equilibrium has been attained, the condensate from both the test and compensating sections is collected and weighed over a measured interval of time. During a run, the condensate levels in both receivers are kept constant by regulating the rate of condensate discharge; and the temperatures of the pipe, the insulation, and the surrounding air are determined occasionally. To calculate the efficiency of the insulation, it is necessary to measure the heat loss from both the bare and the covered surface.

TYPICAL EXPERIMENT

Objects. To determine the:

1. Thermal efficiency of pipe covering.
2. Thermal conductivity of pipe covering.
3. Values of the surface-film coefficients.

Procedure. 1. Install a three-foot section of pipe covering and attach a thermocouple to its surface.

2. Open condensate valves *B* and *C* and pass steam through the system until it emerges relatively dry. Then close these valves and adjust steam valve *A* to give the desired pressure, as indicated by the manometer.

As condensate collects, open valves *B* and *C* slightly to permit the condensate to drain at a rate sufficient to maintain constant levels in the gage glasses.

3. Read the thermocouples on the pipe and insulation and the thermometers in the surrounding air space occasionally until they indicate that equilibrium has been attained; this will require about 3 hours depending upon the type of insulation being tested.

4. After conditions have become substantially constant, make a test run for about 3 to 4 hours, collecting the condensate from the test and compensating sections in beakers. During the run, read the thermocouples and thermometers occasionally, and weigh the condensate about every 30 minutes. If the weight of condensate for any particular period differs too greatly from the average during the other periods, discard the data for that period.

5. Remove the insulation, and after equilibrium conditions have again been attained, repeat the experiment with the bare pipe alone.

6. Calculate:

- a. The thermal efficiency of the insulation.

b. The thermal conductivity of the insulation.

c. The surface-film coefficients for the insulation and the bare pipe.

Typical Data. The experimental data and the calculated results of the experiment are given in Tables 2 to 4; and the results are summarized in Table 5.

TABLE 2
Dimensions of Pipe in Test Section

Symbol	Item	Value
—	Length of test section, ft.	3.0
d_1	Outside diameter of pipe, in.	1.315
A_1	Surface area of test section pipe, ft. ²	1.03
l	Thickness of insulation, in.	0.875
A_2	Surface area of insulation, ft. ²	2.41
A_m	Logarithmic-mean area of insulation, ft. ²	1.62

TABLE 3
Experimental Heat Transfer Data with Insulated Pipe

Symbol	Item	Value
W_1	Weight of condensate from test section, grams/hr.	89.6
W_2	Weight of condensate from compensating section, grams/hr.	48.8
—	Steam pressure, in. of Hg (gage).	2.5
t_1	Temperature of pipe surface, °F.	216
t_2	Temperature of insulation surface, °F.	101
t_a	Temperature of air, °F.	87.5
—	Quality of steam, per cent.	99
λ	Latent heat of steam at 2.5 in. Hg gage, B.t.u./lb.	968

Calculations.

Item 1:* *Heat saved due to insulation, per cent*

$$\begin{aligned}
 &= \frac{(W_3 - W_4) - (W_1 - W_2)}{(W_3 - W_4)} \times 100 \\
 &= \frac{(206.6 - 46.6) - (89.6 - 48.8)}{(206.6 - 46.6)} \times 100 = 74.5
 \end{aligned}$$

Item 2: *Thermal efficiency of insulation.*

The so-called efficiency of pipe covering is the per cent of heat loss from the bare pipe that is saved by covering the pipe with insulation, assuming that the room temperature is the

* Item numbers refer to quantities in Table 5; symbols refer to quantities in Tables 2 to 4.

same in both cases. It should be noted, however, that in this experiment the temperature rose from 87.5°F. to 95.5°F. when the insulation was removed. A correction for this change in temperature can be made if it is assumed that the loss of heat from the bare pipe varies with the temperature in the same ratio as the heat loss from the compensating section.

TABLE 4
Experimental Heat Transfer Data with Bare Pipe

Symbol	Item	Value
W_3	Weight of condensate from test section, grams/hr.	206.6
W_4	Weight of condensate from compensating section, grams/hr.	46.6
—	Steam pressure, in. of Hg (gage).	2.5
t_s	Temperature of pipe surface, °F.	216
t_a	Temperature of air, °F.	95.5
—	Quality of steam, per cent.	99
λ	Latent heat of steam at 2.5 in. Hg gage, B.t.u./lb.	968

TABLE 5
Summary of Calculated Results of Heat Transfer Experiment

Item No.	Item	Value
1	Heat saved due to insulation, per cent.	74.5
2	Thermal efficiency of insulation, per cent.	75.5
3	Surface film coefficient $(h_c + h_r)_1$ for bare pipe, with room at 95.5°F., B.t.u./(hr. \times ft. ² \times °F.).	2.72
4	Surface film coefficient $(h_c + h_r)_2$ for insulation, with room at 87.5°F., B.t.u./(hr. \times ft. ² \times °F.).	2.65
5	Thermal conductivity of insulation, k_1 , B.t.u.-ft./(hr. \times ft. ² \times °F.).	0.034
	B.t.u.-in./(hr. \times ft. ² \times °F.).	0.408
6	Thermal conductivity of insulation based on area of pipe and overall temperature drop from pipe surface to air, k_2 , B.t.u.-ft./(hr. \times ft. ² \times °F.).	0.0475
	B.t.u.-in./(hr. \times ft. ² \times °F.).	0.570

On the basis of this assumption, the quantity of condensate from the bare pipe—had there been no rise in the temperature of the air—would have been:

$$\frac{W_3}{W_4} \times (W_3 - W_4) = \frac{48.8}{46.6} \times (206.6 - 46.6) = 168 \text{ grams/hr.}$$

Therefore, the thermal efficiency is

$$\frac{168 \times (W_1 - W_2)}{168} \times 100 = \frac{168 \times (89.6 - 48.8)}{168} \times 100$$

$$= 75.5 \text{ per cent.}$$

Item 3: *Surface-film coefficient for bare pipe, $(h_e + h_r)_1$.*

The heat loss from bare pipe is given by the equation²

$$\frac{Q_1}{\theta} = (h_e + h_r)_1 A_1 \Delta t_1 \quad (1)$$

Therefore,

$$(h_e + h_r)_1 = \frac{Q_1/\theta}{A_1(t_4 - t_6)} \quad (2)$$

$$= \frac{338}{1.03 \times 120.5} = 2.72 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}),$$

when the air is at a temperature of 95.5°F.

where

$$338 = \frac{Q_1}{\theta} = \text{heat loss from bare pipe, B.t.u.}/\text{hr.}$$

$$= \frac{(W_3 - W_4)}{\text{grams}/\text{lb.}} \times \lambda \times \frac{\text{steam quality}}{100}$$

$$= \frac{(206.6 - 46.6)}{454} \times 968 \times \frac{99}{100}$$

$$1.03 = A_1 = \text{surface area of pipe, ft.}^2$$

$$120.5 = t_4 - t_6 = \text{temperature difference between pipe surface and air, } ^\circ\text{F.}$$

$$= 216 - 95.5$$

Item 4: *Surface-film coefficient for insulation, $(h_e + h_r)_2$.*

$$(h_e + h_r)_2 = \frac{Q_2/\theta}{A_2(t_2 - t_3)} = \frac{86.1}{2.41 \times 13.5}$$

$$= 2.65 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}), \text{ when the air is at a temperature of } 87.5^\circ\text{F.}$$

where

$$86.1 = \frac{Q_2}{\theta} = \text{heat loss from insulated pipe, B.t.u.}/\text{hr.}$$

$$\begin{aligned}
 &= \frac{W_1 - W_2}{\text{grams/lb.}} \times \lambda \times \frac{\text{quality of steam}}{100} \\
 &= \frac{89.6 - 48.8}{454} \times 968 \times \frac{99}{100} = 86.1 \text{ B.t.u./hr.}
 \end{aligned}$$

2.41 = A_2 = surface area of insulation, ft.²

13.5 = $t_2 - t_4$ = temperature difference between insulation surface and air, °F.

$$= 101 - 87.5$$

Item 5: *Thermal conductivity of the insulation, k_1 .*

The heat loss through the insulation is given by the equation

$$\frac{Q_1}{\theta} = \frac{k_1 A_m (t_1 - t_2)}{l/12} \quad (3)$$

Therefore,

$$\begin{aligned}
 k_1 &= \frac{(Q_1/\theta) \times l/12}{A_m (t_1 - t_2)} \quad (4) \\
 &= \frac{86.1 \times 0.0729}{1.62 \times 115} \\
 &= 0.034 \text{ B.t.u.-ft.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}) \\
 &= 0.034 \times 12 = 0.408 \text{ B.t.u.-in.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}).
 \end{aligned}$$

where

$$86.1 = \frac{Q_1}{\theta} = \text{heat loss, B.t.u./hr.}$$

1.62 = A_m = log-mean area of the pipe covering, ft.²

115 = $t_1 - t_2$ = temperature difference between pipe surface and insulation surface, °F.

$$= 216 - 101$$

$$0.0729 = l/12 = \text{thickness of pipe covering, ft.} = \frac{0.875}{12}$$

Item 6: *Thermal conductivity of the insulation, based on area of pipe and temperature drop from pipe surface to air, k_2 .*

The calculation of the thermal conductivity in Item 5 is based on the assumption that the contact between the insulation and the pipe is sufficiently good so that there is no appre-

ciable temperature drop between the surface of the pipe and the inner surface of the insulation. If an appreciable temperature drop exists across the pipe-insulation boundary, the drop across the insulation itself will be correspondingly lower and the value of k will be proportionally higher. Because of the difficulty in estimating the temperature drop across this boundary, and the lack of accurate data for the values of $h_c + h_r$ for covered surfaces, many manufacturers report the value of k in B.t.u. per degree temperature difference from pipe to air per square foot of pipe surface.

Calculated on this basis:

$$\begin{aligned} k_1 &= \frac{(Q_1/\theta) \times l/12}{A_1(t_1 - t_3)} = \frac{86.1 \times 0.875/12}{1.03(216 - 87.5)} \\ &= 0.0475 \text{ B.t.u.-ft.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}) \\ &= 0.0475 \times 12 = 0.570 \text{ B.t.u.-in.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}). \end{aligned}$$

Discussion of Results. 1. Under the same conditions of steam pressure, steam quality, and room temperature, efficiency tests made on the same sample of pipe covering check within one per cent.

Tests on samples of the same type of pipe covering obtained from a manufacturer at different times give results differing by more than one per cent. This is probably due to slight differences in the composition of the manufactured product.

2. From Perry's *Chemical Engineers' Handbook*³, a value of 2.66 B.t.u./(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}) is obtained for the combined coefficient of convection and radiation ($h_c + h_r$) for bare pipe at a temperature of 216°F. when the room temperature is 81°F. The value obtained in the present experiment was 2.74 B.t.u./(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}) with a room temperature of 95.5°F. Considering the difference in room temperature and possible differences in air convection, the two values check reasonably closely.

3. The value obtained for the film coefficient ($h_c + h_r$) for the insulation was 2.65 B.t.u./(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}). Although such data are rather meager for insulated surfaces, this value is probably somewhat high. If a value of 2.0 is assumed for ($h_c + h_r$) and the thermal conductivity k_1 of the insulation is calculated from the equation

$$\frac{Q_1}{\theta} = \frac{(t_1 - t_2)}{\frac{l/12}{k_1 A_m} + \frac{1}{(h_c + h_r)_1 A_1}} \quad (5)$$

a value of 0.035 B.t.u.-ft./ (hr. \times ft.² \times °F.) is obtained. This corresponds to the value of 0.034 obtained experimentally.

4. The manufacturer of the sample used in this test suggests a loss of 0.672 B.t.u. per hour per foot length of 1 inch pipe per °F. temperature difference from pipe to room at 70°F. as an average value for pipe covering of this type.

If it is assumed that the overall heat loss per degree does not change with temperature, the calculated heat loss from the insulation when the room temperature is 87.5°F. would be

$$0.672 \times (216 - 87.5) \times 1.03 = 88.9 \text{ B.t.u./hr.}$$

and the thermal conductivity would be

$$k_2 = \frac{88.9 \times (0.875/12)}{1.03 \times (216 - 87.5)} = 0.049 \text{ B.t.u.-ft./ (hr. } \times \text{ ft.}^2 \times \text{ °F.)}$$

$$= 0.049 \times 12 = 0.588 \text{ B.t.u.-in./ (hr. } \times \text{ ft.}^2 \times \text{ °F.)}$$

Although these calculations were based on the assumption that the overall heat loss per degree at 87.5°F. would be the same as at 70°F.—an assumption that is not strictly correct—the calculated value of the thermal conductivity and the corresponding experimentally-determined value—0.570 B.t.u.-in./ (hr. \times ft.² \times °F.)—agree closely enough for industrial purposes.

NOMENCLATURE

- A = surface area, ft.²
- A_m = logarithmic-mean area, ft.²
- d = diameter, in.
- $h_c + h_r$ = film coefficient of heat transfer by conduction and convection, and radiation, B.t.u./ (hr. \times ft.² \times °F.).
- k = thermal conductivity, B.t.u.-ft./ (hr. \times ft.² \times °F.).
- l = thickness of insulation, in.
- t = temperature, °F.
- Δt = temperature difference, °F.
- Q/θ = heat transferred, B.t.u./hr.
- W = weight of condensate, grams/hr.
- λ = latent heat of steam, B.t.u./lb.

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2. PERRY, "Chemical Engineers' Handbook," McGraw-Hill Book Company, New York, 1941, 2nd ed., pp. 985-7.
3. *Ibid.*, p. 987.

AN APPARATUS FOR DETERMINING HEAT LOSS FROM BARE AND LAGGED PIPES

Designed and Constructed by
The Department of Chemical Engineering
University of Rochester*

Description. In this apparatus** for determining the heat loss from bare and lagged pipe, heat is supplied by the dead condensation of steam, and surface temperatures are measured by means of a portable thermocouple potentiometer. The test section consists of four 10-foot lengths of 1-inch standard steel pipe mounted, with a slope of 0.63 inch per foot, on a framework of welded 2-inch steel angles; one of the lengths is bare, and the others are covered with *air-cell* asbestos insula-

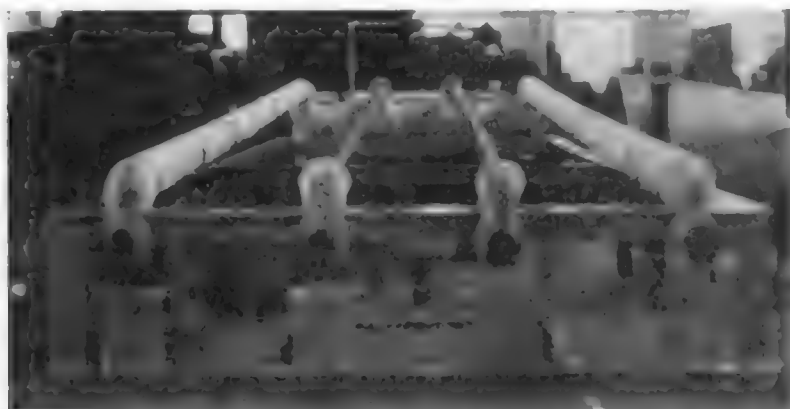


Figure 1. View of Apparatus for Determining Heat Loss from Bare and Lagged Pipes—University of Rochester

tion, 85 per cent magnesia insulation, and *silver chrome* (aluminum pigment) paint, respectively.

The test pipes are connected to a common header into which steam is introduced either directly from the main or from a line containing a reducing valve. The steam condensate is drained from the opposite ends of the pipes through plug-type valves, and is collected in beakers and measured.

To minimize heat loss from the system, all parts of the apparatus, with the exception of the test pipes themselves, are covered with 85 per cent magnesia insulating cement.

Figure 1 is a photograph of the apparatus from the condensate dis-

* Reported by H. S. Gardner.

** This apparatus was designed and constructed by Roger Drexel and Robert Rothfus as part of a senior chemical engineering project.

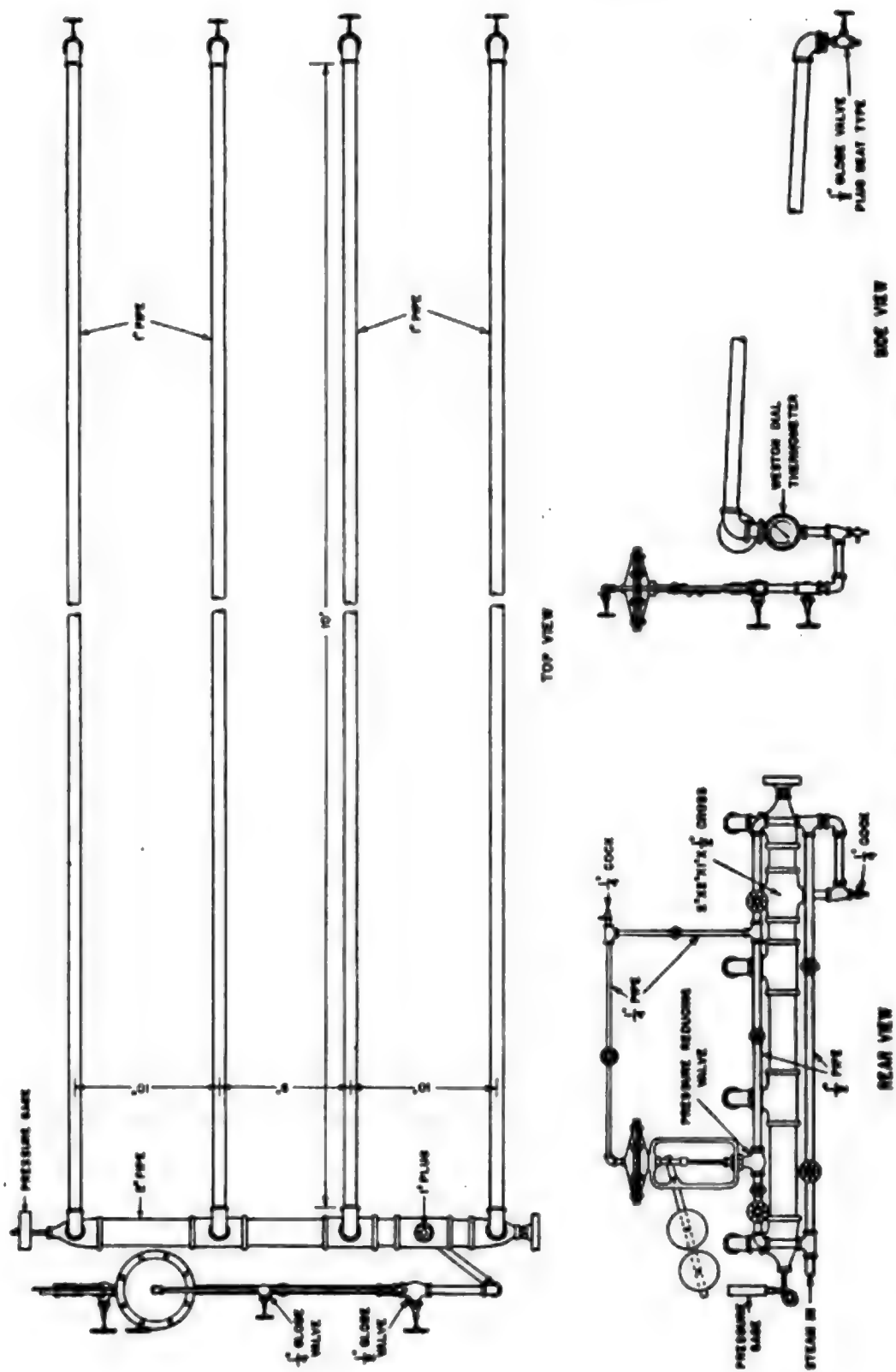


Figure 2. Apparatus for Determining Heat Loss from Bare and Lagged Pipes—University of Rochester

charge end, and Figure 2 is a drawing of the assembly, exclusive of the framework.

TABLE 1

Bill of Materials for Apparatus for Determining Heat Loss from Bare and Lagged Pipes

University of Rochester

Quantity	Description	Size	Material
2 ft.	Std. pipe	2 in.	Black wrought steel
40 ft.	" "	1 in.	" " "
6 ft.	" "	$\frac{1}{2}$ in.	" " "
4 ft.	" "	$\frac{1}{2}$ in.	" " "
2	Standard nipple	2 in. x 3 in. long	" " "
4	" "	1 in., short	" " "
1	" "	1 in., close	" " "
5	" "	$\frac{1}{2}$ in. x $2\frac{1}{2}$ in. long	" " "
1	" "	$\frac{1}{2}$ in., short	" " "
2	Standard tee	2 x 2 x 1 in.	Black malleable iron
1	" "	2 x $\frac{1}{2}$ x 1 in.	" " "
2	" "	1 x $\frac{1}{2}$ x $\frac{1}{2}$ in.	" " "
1	" "	$\frac{1}{2}$ x $\frac{1}{2}$ x $\frac{1}{2}$ in.	" " "
1	" "	$\frac{1}{2}$ x $\frac{1}{2}$ x $\frac{1}{2}$ in.	" " "
4	Std. 90° elbow	1 in.	" " "
2	" " "	$\frac{1}{2}$ in.	" " "
2	" " "	$\frac{1}{2}$ in.	" " "
3	Ground-joint union	$\frac{1}{2}$ in.	" " "
1	" " "	$\frac{1}{2}$ in.	" " "
1	Cross	2 x 2 x 1 x $\frac{1}{2}$ in.	" " "
1	Plug	$\frac{1}{2}$ in.	Cast iron
3	Globe valve (125 lb.)	$\frac{1}{2}$ in.	Brass
1	" " "	$\frac{1}{2}$ in.	"
2	Pet cock	$\frac{1}{2}$ in.	"
4	Plug-type valve (150 lb.)	$\frac{1}{2}$ in.	"
1	Pressure gage	0 to 100 lb., $3\frac{1}{2}$ in. face	"
1	Weston temperature gage, Model 221D	50 to 400 °F.	Stainless steel
1	Pressure-reducing valve	$\frac{1}{2}$ in. (3 to 40 lb.)	Steel; bronze trimmed
80 ft.	Angles	2 x 2 x $\frac{1}{8}$ in.	Steel
12 ft.	Pipe insulation	1 in.	Air-cell
12 ft.	" "	1 in.	85% magnesia

Materials and Cost. The cost of the materials for this apparatus, as listed in Table 1, is approximately \$90. Portable equipment, such as beakers, scales, stop watch, and pyrometer—Illinois Testing Labora-

tories Portable Pyrocon—, are not included in the bill of materials or in the cost estimate because their use is not limited to this piece of equipment.

Operation. In measuring heat losses with this apparatus, steam at the desired pressure is admitted to the header, and the plug-type valves are cracked slightly to permit the condensate to drain as fast as it is formed without allowing more than a minimum of live steam to escape. Temperatures are measured at various points on the surfaces of the test pipes, and when they indicate that equilibrium has been attained the condensate is collected over a timed interval. The surface-film coefficients and the lagging efficiency are calculated from the temperatures of the surfaces and the room, the quantity of condensate, and the pressure of the steam.

TYPICAL EXPERIMENT

Objects. 1. To determine the combined radiation and convection coefficient ($h_c + h_r$) at various temperatures from different surfaces: (a) bare pipe, (b) pipe painted with silver-chrome paint, (c) air-cell insulation, and (d) 85% magnesia insulation.

2. To determine the efficiency of these insulating materials.

3. To compare the experimentally determined values of ($h_c + h_r$) with those calculated from empirical equations.

Procedure. 1. Make three runs with steam at approximately 10 pounds per square inch gage, 40 pounds per square inch gage, and full line pressure.

2. For each run:

a. After adjusting the system to the desired pressure, *crack* the drain cock under the header to remove water from the steam line and header.

b. Open the four plug-type valves to blow out any condensate from the pipes, and then close them until only a small amount of steam escapes along with the condensate.

c. When the system has reached equilibrium, as determined by surface temperature measurements, collect and measure the condensate from each pipe over a timed interval of 15 to 30 minutes, and during this period record the following data:

- (1) Barometric pressure.
- (2) Room temperature.
- (3) Steam pressure and temperature.
- (4) Surface temperatures.

TABLE 2

Experimental Data for the Determination of Heat Loss from Bare and Insulated Pipes

		Length of pipes = 10 ft.											
Pipe No.		1				2				3			
Covering		Air-cell insulation				None				Silver-chrome paint			
Outside diam., in.		2.875				1.315				1.315			
Emissivity		0.95				0.95				0.35			
Run No.		1				2				3			
Barometric pressure, in. of Hg		29.58				29.58				29.58			
Steam press., lb./in. ² gage		10				42				118			
Steam temp., °F.		236				287				345			
Room temp., °F.		77				77				77			
Length of run, min.		26.7				17.1				17.3			
Pipe No.		1	2	3	4	1	2	3	4	1	2	3	4
1st set	a*	110	215	213	110	117	255	253	134	126	316	316	130
	b	115	218	215	117	132	258	261	139	146	317	318	138
	c	118	217	215	125	140	263	269	138	146	316	313	142
	d	118	218	217	125	130	260	267	142	—	—	—	—
	e	120	218	220	125	—	—	—	—	—	—	—	—
Surface temperature readings, °F.	a	108	217	213	110	120	253	256	140	122	314	317	136
	b	112	217	217	117	134	260	263	147	148	317	316	150
	c	118	217	215	122	135	262	261	143	143	318	318	144
	d	118	217	220	123	130	260	259	138	—	—	—	—
	e	119	218	217	120	—	—	—	—	—	—	—	—
3rd set	a	110	215	215	110	120	255	262	149	126	298**	319	147
	b	112	219	216	115	135	254	268	150	143	317	320	159
	c	117	216	220	120	135	265	264	129	142	316	316	136
	d	116	216	217	122	128	265	263	132	—	—	—	—
	e	116	217	220	123	—	—	—	—	—	—	—	—
Average ..		115	217	217	119	130	259	262	139	139	316	317	142
Volume of condensate, ml.		148	351	310	121	116	300	252	96	150	420	358	110

* Positions at which readings were taken.

** Discarded in averaging results.

Surface temperatures should be taken at three or more equally spaced points along each test pipe, and at least three sets of readings should be taken during each run. Because steam rising from the condensate valves tends to heat the pipes and insulation, no temperature measurements should be made within 20 inches of the exit ends of the pipes.

3. From the experimental data for each run, calculate the combined film coefficients for convection and radiation and the lagging efficiencies,

TABLE 3
Summary of Calculations for Heat Loss from Bare and Lagged Pipes
Room temp., $t_r = 77^\circ\text{F}$.

Run No.	Pipe No.	Type of covering	Average surface temp. $^\circ\text{F}$, t_s	Temperature difference, surface to room, $^\circ\text{F}$, Δt	Coefficients from experimental data, B.t.u./ (hr. \times ft. ² \times $^\circ\text{F}$), ($h_c + h_r$)	Lagging efficiency, per cent, L.E.	Coefficients from equations, B.t.u./ (hr. \times ft. ² \times $^\circ\text{F}$)		
							h_c	h_r	($h_c + h_r$)
1	1	Air-cell asbestos	115	38	2.51	58	0.80	1.13	1.93
	2	None	217	140	3.55	—	1.35	1.49	2.84
	3	Silver-chrome paint	217	140	3.14	12	1.35	0.55	1.90
	4	85% magnesia	119	42	1.68	66	0.80	1.16	1.96
2	1	Air-cell asbestos	130	53	2.28	61	0.87	1.18	2.05
	2	None	259	182	3.70	—	1.44	1.62	3.06
	3	Silver-chrome paint	262	185	3.06	16	1.45	0.62	2.07
	4	85% magnesia	139	62	1.43	68	0.88	1.20	2.08
3	1	Air-cell asbestos	139	62	2.50	64	0.91	1.20	2.11
	2	None	316	239	3.98	—	1.54	1.92	3.46
	3	Silver-chrome paint	317	240	3.37	15	1.55	0.71	2.26
	4	85% magnesia	142	65	1.58	74	0.89	1.20	2.09

and compare the experimental values of the coefficients with those calculated from empirical equations.

Typical Results*. The experimental data for three runs are given in Table 2 and a summary of the calculations is given in Table 3. The calculations were made as follows:

The rate of heat loss from the surface may be expressed as:

$$Q/\theta = (h_c + h_r)A(\Delta t) \quad (1)$$

* These results were taken from a student laboratory report.

Therefore,

$$(h_c + h_r) = \frac{Q/\theta}{A(\Delta t)} \quad (2)$$

where

Q/θ = heat transferred, B.t.u./hr.

h_c = film coefficient of heat transfer by convection,
B.t.u./(hr. \times ft.² \times °F.).

h_r = film coefficient of heat transfer by radiation,
B.t.u./(hr. \times ft.² \times °F.).

A = surface area, ft.²

Δt = temperature difference, surface to air, °F.

The quantity Q/θ is calculated from the quantity of steam condensate, the latent heat of vaporization, and the time of the run. However, because the condensate leaves the system at a pressure higher than atmospheric, some of it flashes, and the volume of condensate collected is smaller than the amount of steam condensed. It is necessary, therefore, to calculate the quantity of condensate lost through flashing; this is done by a heat balance:

$$H_{l_1} = X(H_{l_2}) + (1 - X)(H_{v_2}) \quad (3)$$

where

H_{l_1} = enthalpy of liquid water at the temperature corresponding to the pressure in the system, B.t.u./lb.

H_{l_2} = enthalpy of liquid water at the temperature corresponding to atmospheric pressure, B.t.u./lb.

H_{v_2} = enthalpy of water vapor at the temperature corresponding to atmospheric pressure, B.t.u./lb.

X = lb. water collected per lb. steam condensed.

CALCULATIONS FOR PIPE NO. 1, RUN NO. 1

1. Values of $(h_c + h_r)$ from Experimental Data

a. Calculation of Q/θ :

$$\begin{aligned} (1) \text{ Atmospheric pressure} &= 29.58 \text{ in. Hg} \\ &= \frac{29.58}{29.92} \times 14.7 = 14.5 \text{ lb./in.}^2 \text{ abs.} \end{aligned}$$

$$\begin{aligned} (2) \text{ Pressure in system} &= 10 \text{ lb./in.}^2 \text{ gage} \\ &= 10 + 14.5 = 24.5 \text{ lb./in.}^2 \text{ abs.} \end{aligned}$$

$$(3) H_{l_1} = 207.2 \text{ B.t.u./lb. at } 24.5 \text{ lb./in.}^2 \text{ abs.}$$

$$(4) H_{i_1} = 179.3 \text{ B.t.u./lb. at } 14.5 \text{ lb./in.}^2 \text{ abs.}$$

$$(5) H_{v_1} = 1150.0 \text{ B.t.u./lb. at } 14.5 \text{ lb./in.}^2 \text{ abs.}$$

Substituting in Equation (3):

$$207.2 = X(179.3) + (1 - X) 1150.0$$

and

$$X = 0.970 \text{ lb. water collected per lb. steam condensed.}$$

Therefore,

$$Q/\theta = \frac{148 \times 1.0}{454 \times 0.970} \times \frac{60}{26.7} \times 953 = 720 \text{ B.t.u./hr.}$$

where

148 = ml. condensate collected in 26.7 min.

1.0 = density of water, gram/ml.

454 = grams/lb.

953 = latent heat of vaporization at 24.5 lb./in.² abs.

b. *Calculation of Area:*

$$A = \frac{\pi(2.875)}{12} \times 10 = 7.54 \text{ ft.}^2$$

where

2.875 = outside diameter of insulation, in.

10 = length of pipe, ft.

c. *Calculation of Δt :*

$$\Delta t = 115 - 77 = 38^\circ\text{F.}$$

where

115 = average surface temperature (Table 2).

77 = room temperature (Table 2).

Substituting the values of Q/θ , A and Δt in Equation (2):

$$h_c + h_r = \frac{720}{7.54 \times 38} = 2.51 \text{ B.t.u./hr.} \times \text{ft.}^2 \times ^\circ\text{F.}.$$

2. Lagging Efficiency

The per cent lagging efficiency is expressed as

$$\text{L.E.} = \frac{q_s - q_L}{q_s} \times 100 \quad (4)$$

where

q_B = heat loss from bare pipe.

q_L = heat loss from lagged pipe.

Since the heat loss is proportional to the quantity of condensate collected, Equation (4) can be written as

$$\text{L.E.} = \frac{w_B - w_L}{w_B} \times 100 \quad (5)$$

where w_B and w_L are the quantities of condensate from the bare and lagged pipes, respectively.

Therefore, for Pipe No. 1, Run No. 1, the lagging efficiency is

$$\text{L.E.} = \frac{351 - 148}{351} \times 100 = 58\%.$$

where 351 and 148 are ml. of condensate from the bare and insulated pipe, respectively (Table 2).

3. Values of $(h_c + h_r)$ from Equations

The equations used for these calculations are:

$$h_c = 0.42 \left(\frac{\Delta t}{D'} \right)^{0.25} \quad (6)$$

and

$$h_r = \frac{0.173p \left[\left(\frac{T_s}{100} \right)^4 - \left(\frac{T_r}{100} \right)^4 \right]}{\Delta t} \quad (7)$$

where for Pipe No. 1, Run No. 1

Δt = temperature difference, surface to air = 38°F.

T_s = surface temperature = 77 + 460 = 537°R.

T_r = room temperature = 115 + 460 = 575°R.

D' = outside diameter = 2.875 in.

p = emissivity = 0.95.

Therefore,

$$h_c = 0.42 \left(\frac{38}{2.875} \right)^{0.25} = 0.80 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.})$$

and

$$\begin{aligned} h_r &= \frac{0.173 \times 0.95 \left[\left(\frac{575}{100} \right)^4 - \left(\frac{537}{100} \right)^4 \right]}{38} \\ &= 1.13 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}) \end{aligned}$$

and

$$h_c + h_r = 0.80 + 1.13 = 1.93 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}).$$

Comments on Results. The experimental results agree reasonably well with those calculated from the equations, although, with the exception of the results for the magnesia covered pipe, the experimental results are higher than the corresponding calculated values. This may have been due to one or more of the following factors: (1) the unknown quality of the steam; (2) condensation in portions of the apparatus other than the test pipes; (3) condensation in the collecting beakers of some of the steam that escaped with the condensate; (4) convection currents in the air; and (5) the fact that the test pipes were not perfectly horizontal.

1. *Quality of the Steam.* In the calculations, the quality of the steam was taken as 100 per cent; actually it was probably about 96 to 98 per cent and the quantity of condensate, therefore, was from 2 to 4 per cent too high. However, this would account for only a relatively small part of the total discrepancy.

2. *Condensation of Steam in Parts of the System Other Than the Test Pipes.* At the end of each test pipe there is a small section which is not included in the 10 foot test lengths; some steam condenses in these sections, particularly at the lower ends where the discharge valves are located. Although this condensation is probably not very great, it will affect the insulated pipes to a relatively greater degree than the bare or painted pipes because, although the ends are insulated, magnesia insulating cement is less efficient than the magnesia pipe covering.

3. *Condensation in Collecting Beakers.* In calculating the quantity of heat transferred, it is assumed that any steam passing out with the condensate, and vapors resulting from the flashing of the condensate will pass off to the atmosphere. However, some condensation undoubtedly occurs in the short nipples on the discharge sides of the valves and also in the collecting beakers. For this reason, it is desirable that the beakers be heated before the test run is started, and that they be insulated. Furthermore, it is essential that only a very small quantity of steam be permitted to escape with the condensate.

4. *Convection Currents in the Air.* This is probably the most serious source of error involved. As the apparatus was not shielded, there was probably considerable movement of air over the pipes, and this tends to greatly increase heat loss from the surfaces.

5. *Slope of the Pipes.* The heat loss by convection was calculated from the equation for horizontal pipes, but the pipes in the experiment

were at a slight angle to the horizontal; it is doubtful, however, if this would have much effect on the results.

In contrast to these errors, which tend to give higher results for the experimental coefficients, there is one error that operates in the opposite direction: this is re-evaporation of condensate in the beakers. As the condensate is maintained at a temperature very near its boiling point, this error may be considerable, particularly where the quantity of condensate is relatively small. This may have accounted for the lower results for the pipes insulated with 85 per cent magnesia.

NOMENCLATURE

- A = surface area, ft.²
- D' = outside diameter, in.
- h_c = film coefficient of heat transfer by conduction and convection, B.t.u./ (hr. \times ft.² \times °F.).
- h_r = film coefficient of heat transfer by radiation, B.t.u./ (hr. \times ft.² \times °F.).
- H_{l_1} = enthalpy of liquid water at the temperature corresponding to the pressure in the system, B.t.u./lb.
- H_{l_2} = enthalpy of liquid water at the temperature corresponding to atmospheric pressure, B.t.u./lb.
- H_v = enthalpy of water vapor at the temperature corresponding to atmospheric pressure, B.t.u./lb.
- L.E. = lagging efficiency, per cent.
- p = emissivity.
- q_b = heat loss from bare pipe.
- q_L = heat loss from lagged pipe.
- Q = heat transferred, B.t.u.
- T_s = surface temperature, °R.
- T_r = room temperature, °R.
- Δt = temperature difference, surface to air, °F.
- w_b = condensate from bare pipe, ml.
- w_L = condensate from lagged pipe, ml.
- X = lb. water collected per lb. steam condensed.
- θ = time, hr.

A DRIER AND TEMPERING COIL

Designed and Constructed by

The Department of Chemical Engineering
Drexel Institute of Technology*

Description. This apparatus, which can be used for measurements involving drying, heat transfer, and fluid flow, consists essentially of: (1) a drying compartment in which a platform is suspended from the beam of a balance resting on top of the drier, (2) a tempering coil made of a bank of vertical, steam-heated copper tubes that provide a heating surface of 17.4 square feet, (3) a small Sirocco blower which supplies the air to the unit, and (4) a duct system that includes tapered transition pieces on the inlet and exit sides of the drier. The transition pieces eliminate undue turbulence in the air stream and distribute the air uniformly over the cross section of the drying compartment.

Figure 1 is a view of the assembled unit. The details of construction of the tempering coil, the framework of the drier, the tapered end-transition pieces, and the door are shown in Figures 2 to 5.

Materials and Cost. The materials used in the construction of this drier and tempering coil are given in Table 1; their cost amounted to approximately \$110.

Construction of the Drier. 1. The *tempering coil*, or heating unit (Figure 2), consists of a vertical bank of 45 copper tubes, 1-inch nominal diameter and 19 inches long, with steam chests at the top and bottom. The tubes are arranged on vertices of equilateral triangles, 2 inches on a side. This results in a bank of 6 rows, with 7 and 8 tubes, respectively, in alternate rows.

2. The *steam chests* are made from $\frac{1}{4}$ -inch steel plate and $3 \times 1\frac{1}{2} \times \frac{1}{8}$ -inch steel channels. Four sheets of the steel plate, each 14 x 18 inches, are cut for the tube sheets and the steam chest covers. The two tube sheets are clamped together, and the holes for the 45 tubes are laid out, drilled, and reamed to $1\frac{1}{8}$ -inches. Two pieces of channel, each 68 inches in length, are cut and formed into frames for the steam chests. The frames are made by cutting miters in the webs of the channels at points corresponding to the corners of the steam chest, heating the channels at these points, bending them to shape, and arc welding the corners. The welds are then ground and filed smooth.

3. The cover plates are clamped in place, and twenty-eight $\frac{3}{8}$ -inch

* Reported by H. T. Ward.

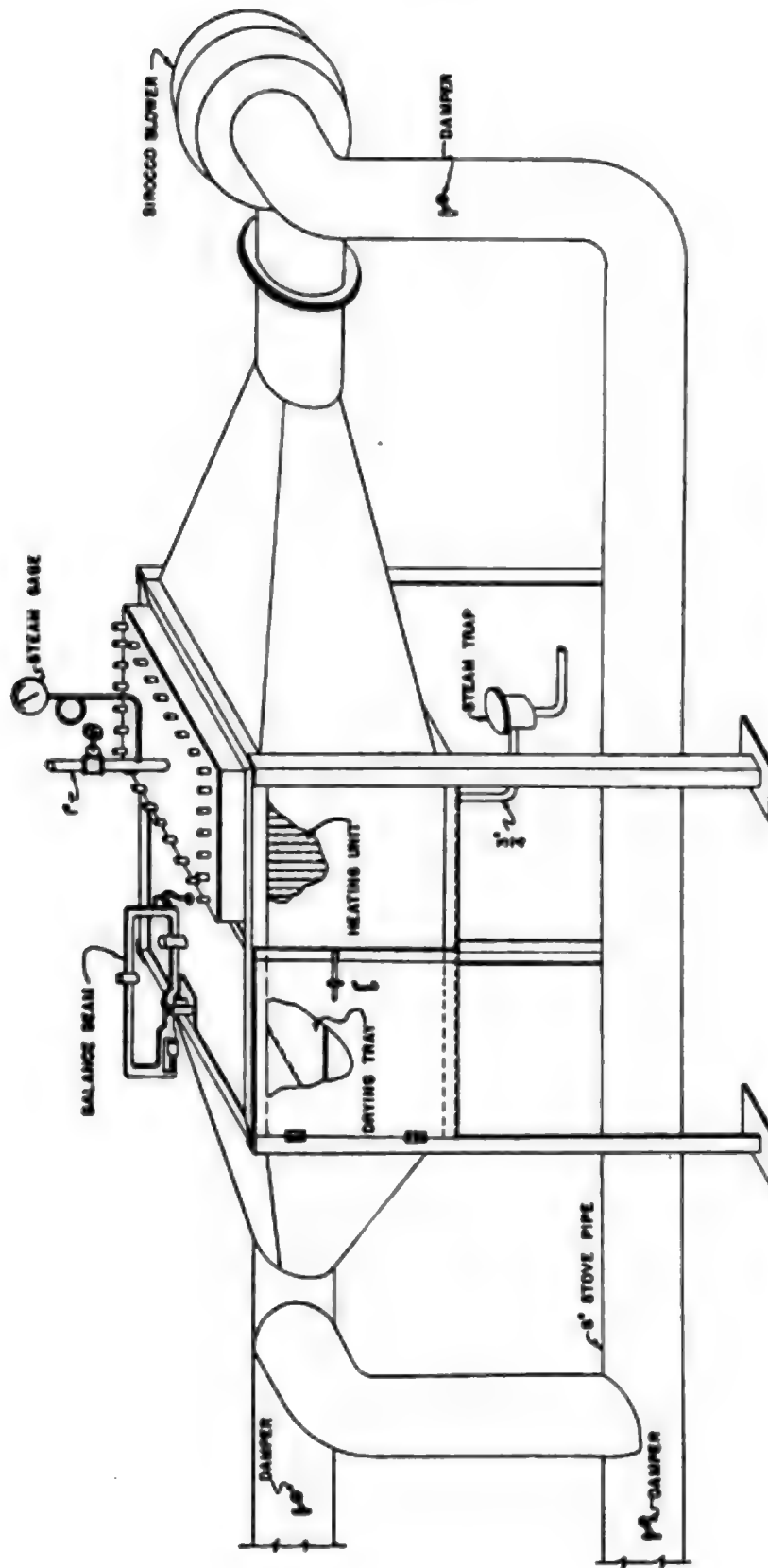


Figure 1. Assembly of Drier and Tempering Coil—Drexel Institute of Technology

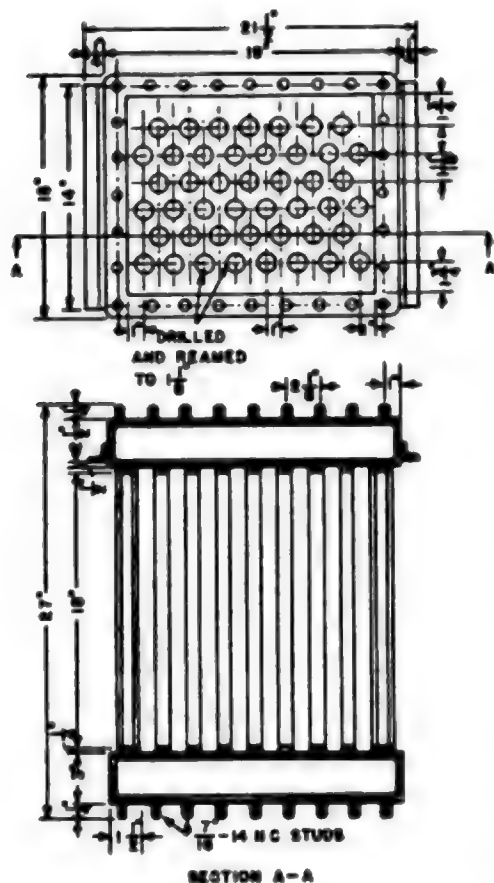


Fig. 2

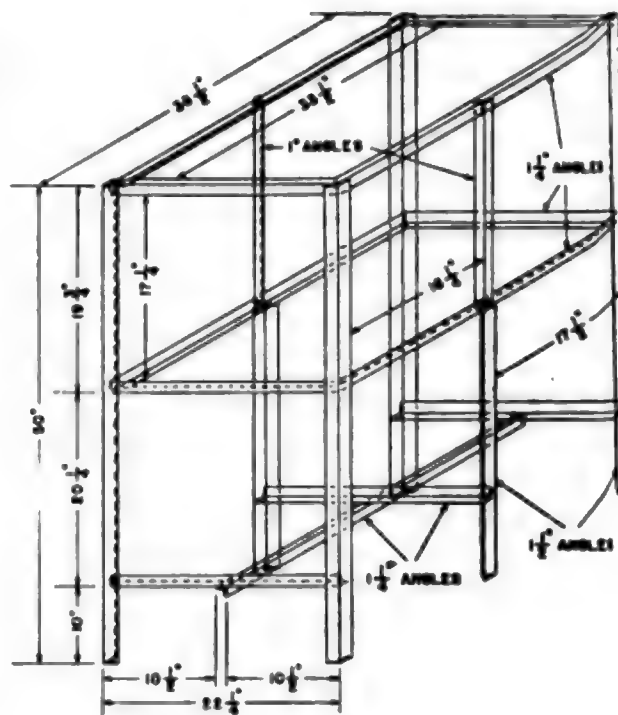


Fig. 3

Figure 2. Details of Tempering Coil
Figure 3. Details of Framework

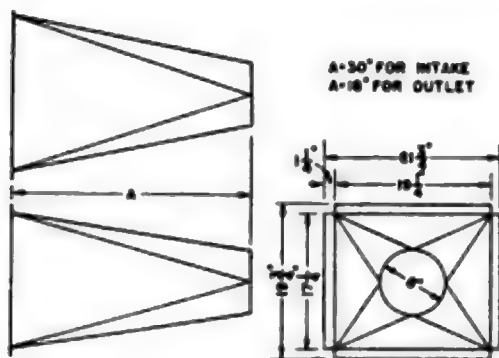


Fig. 4

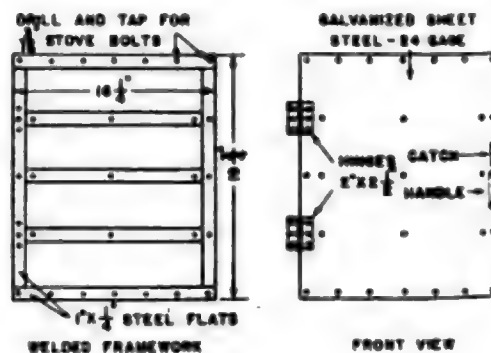


Fig. 5

Figure 4. Details of End Transition Pieces
Figure 5. Details of Door

holes are drilled on $2\frac{1}{2}$ -inch centers. The plates are then removed, the holes in the frame are tapped for $\frac{1}{8}$ inch-14 NC threads, and the studs are threaded in and welded fast to keep them from turning and to prevent leaks.

TABLE 1
Bill of Materials for Drier and Tempering Coil
Drexel Institute of Technology

Quantity	Description	Size	Material
2 ft.	Std. pipe	1 in.	Black wrought steel
2 ft.	" "	$\frac{3}{4}$ in.	" " "
80 ft.	Tubing	1 in.	Type K, hard copper
20 ft.	Angles	$1\frac{1}{2} \times 1\frac{1}{2} \times \frac{1}{8}$ in.	Steel
30 ft.	"	$1\frac{1}{2} \times 1\frac{1}{2} \times \frac{1}{8}$ in.	"
15 ft.	"	1 x 1 x $\frac{1}{8}$ in.	"
12 ft.	Channels	3 x $1\frac{1}{2}$ x $\frac{1}{8}$ in.	"
8 sq. ft.	Sheet	$\frac{1}{2}$ in.	"
48 sq. ft.	"	24 gage	Galvanized steel
56	Studs & nuts	$\frac{1}{2}$ in.- 14NC x $1\frac{1}{2}$ in. long	Steel
200	Round-head screws	$\frac{1}{2}$ in.- 20NC	"
1	Std. gate valve	1 in.	Brass
1	Steam gage	0 to 100 lb., $3\frac{1}{2}$ in. face	"
1	Steam-gage siphon	$\frac{1}{2}$ in.	Black wrought steel
1	Steam trap	$\frac{1}{2}$ in.	Cast iron
24 ft.	Stove pipe	8 in.	Galvanized steel
3	Stove-pipe elbows	8 in.	" "
1	" " tee	8 in.	" "
3	" " dampers	8 in.	Cast iron
9 ft.	Strap	1 x $\frac{1}{2}$ in.	Steel
2	Hinges	$2\frac{1}{2} \times 2$ in.	"
1	Door pull	$7\frac{1}{2} \times 1$ in.	"
1	Door latch	—	"
1	Sirocco blower & motor	500 cu. ft. of free air per min.	—
1	Double-beam balance	1 kilogram capacity	—
6 sq. ft.	Gasket material	$\frac{1}{8}$ in. thick	Garlock # 900

4. The stud holes in the cover plates are enlarged to $\frac{1}{2}$ -inch, holes are drilled and tapped in the center of each plate to provide openings for the 1-inch steam inlet and the $\frac{1}{2}$ -inch condensate outlet, and two braces of 1-inch angles are welded to each cover plate to prevent them from bulging under heat and pressure. Nipples are then threaded into the center openings and welded on both sides of the plates.

5. Four 18-inch lengths of 1-inch angles are cut for use as spacers for the tube sheets. These angles are welded to the sheets, and the sheets are then arc welded to the frames, or sides, of the steam chests.

6. Brackets of $1\frac{1}{4}$ -inch angles are welded to the front and back of the lower steam chest to support the heating unit in the drier.

7. The copper tubing is cut into 19-inch lengths, and these are driven into the tube sheets and expanded in place with a $1\frac{1}{4}$ -inch tube-expanding tool. Tubes of this length project about $\frac{1}{4}$ -inch through each tube sheet, and they can be beaded if desired; however, as they are under very little lateral stress, this operation is unnecessary.

8. Gaskets, cut from *Garlock #900* gasket material, are fitted in place, and the cover plates are then bolted to the frames of the steam chest. Finally, the unit is tested for leaks with water at a pressure of 50 pounds per square inch.

9. The *frame of the drier* is made from $1\frac{1}{4}$ -inch and $1\frac{1}{2}$ -inch steel angles, arc welded together as shown in Figure 3. Before the frame is assembled, the angles are drilled and tapped to take $\frac{1}{2}$ inch-20 NC round head screws, which are used to attach the sheet metal covering.

10. The *air duct* is 8-inch galvanized stove pipe; and connections between this pipe and the drier are made through tapered, transition pieces (Figure 4) cut from a 36 x 19-inch sheet of 24-gage galvanized steel and bent and soldered together as shown. In laying out these sections, it is most convenient to first make patterns from heavy paper.

11. The heating coil is fitted in place, and the transition pieces and the 24-gage sheet metal covering are bolted to the framework.

12. The *door* is made from 1 x $\frac{1}{4}$ -inch steel flats and galvanized sheet steel, as shown in Figure 5. The flats are cut and welded together, holes are drilled and threaded for $\frac{1}{4}$ -inch bolts, the frame is covered with the sheet steel, and the hinges, latch, and handle are bolted into position.

13. A small opening is drilled in the top of the drying compartment. The beam of a platform balance is mounted on top of the drier and connected to a platform in the drier by means of a wire passing through this opening. This arrangement is shown in Figure 1.

Operation. Although separate experiments on fluid flow, heat flow, and drying can be performed with this apparatus, it is easier to run one experiment involving all three of these operations, for data necessary for all calculations can be obtained at the same time.

In carrying out an experiment, the operations are essentially as follows:

1. The blower is turned on and the damper in the inlet duct is adjusted to give a satisfactory air velocity.
2. Steam is introduced into the heating unit and, after equilibrium has been attained, a pan of material to be dried is placed on the balance platform.
3. At intervals of time, the pan is weighed, the steam condensate is cooled and weighed, the temperature of the air is measured before and after it passes over the heating unit, the humidity of the entering and exit air is determined, and the air velocity is measured with a pitot tube at various points over the cross section of the fluid stream, or by means of an anemometer in the exit duct.

Auxiliary Equipment. In order to carry out an experiment in this manner, the following auxiliary equipment is needed:

1. Wet- and dry-bulb thermometers, for measuring the humidity of the entering and exit air.
2. A cooler, a receiver, and a platform scale, for measuring the quantity of condensate.
3. A pitot tube or an anemometer, for measuring the air velocity.

TYPICAL EXPERIMENT

Objects. 1. To determine the overall coefficient of heat transfer of a steam-heated tempering coil.

2. To compare the observed coefficients with those calculated from film-coefficient equations found in the literature.

Procedure. 1. Before beginning the experimental work, submit a *preliminary report* giving the answers to the following:

- a. State the equation for the air-film coefficient in a heating unit with air at right angles to a bank of heated tubes. What name is given this equation?
- b. On what area is the mass velocity based in this equation?
- c. What physical data concerning the air must be known in using this equation?
- d. What will be the probable magnitudes of the metal resistance and the steam-film resistance as compared to the air-film resistance?

2. Carry out the experiment in the following manner:

- a. Determine the area of the tempering coil.
- b. Regulate the pressure of the steam to the coil at the desired value.
- c. Pass air at the desired velocity over the tubes.

- d. After conditions have become constant, take and record the following data at frequent intervals:
 - (1) Temperatures of the air streams.
 - (2) Velocity of the air.
 - (3) Steam pressure.
 - (4) Quantity of steam condensate.
 - e. Repeat the experiment at several different air velocities and different steam pressures.
 - f. For each steam pressure, determine the radiation loss from the unit by making a blank run without passing air over the tubes.
3. Include the following in the report of the experimental work:
- a. Calculations for: (1) the overall heat-transfer coefficient based on the experimental data, and (2) the overall heat-transfer coefficient based on film-coefficient equations found in the literature.
 - b. A comparison of the experimental and calculated values of the overall heat-transfer coefficients.
 - c. A discussion of the results, including any comments or conclusions.

Preliminary Report. 1. The *Reiher equation*¹ is used to evaluate the air-film coefficient in a heating unit with air at right angles to a bank of heated tubes. This equation is:

$$h = 0.131 \left(\frac{k_f}{D} \right) \left(\frac{Dv_{\max}}{\mu_f} \right)^{0.7}$$

The values of the thermal conductivity k_f and viscosity μ_f are evaluated at the mean film temperature.

2. The maximum mass velocity v_{\max} is calculated on the area of the minimum opening between adjacent pipes.
3. It is necessary to know the density, viscosity, and specific heat of air in using the Reiher equation.
4. The magnitudes of the metal and steam-film resistances are small compared to the air-film resistance.

Experimental Data and Calculated Results. The experimental data for two runs for the determination of the heat-transfer coefficients of the tempering coil are given in Table 2, and a summary of the calculated results is given in Table 3.

TABLE 2

Experimental Results for Heat Transfer in a Tempering Coil

Symbol	Item	Run No.	
		1	2
P_1	Pressure of exit air, in. of Hg	30.19	30.19
P_2	Average steam pressure, lb./in. ² , gage	23.2	15.1
t_1	Temperature of steam, °F.	264	250
t_2	Average temp. of entering air, °F.	74.2	76.3
t_3	Average temp. exit air, °F.	127.6	118
V	Velocity of air in exit duct, ft./min.	674.2	434.5
W_1	Steam condensed during test run, lb./hr.	15.4	8.62
W_0	Steam condensed during blank run, lb./hr.	6.50	6.43

TABLE 3

Calculated Results for Heat Transfer in a Tempering Coil

Symbol	Item	Run No.	
		1	2
A	Area of tempering coil, ft. ²	19.90	19.90
Δt	Log-mean temperature difference, °F.....	161	152.2
ρ	Density of exit air, lb./ft. ³	0.0683	0.0694
G	Quantity of exit air, lb./hr.....	964	633
	<i>Heat transferred, B.t.u./hr.</i>		
q_1	a. Based on condensate uncorrected for radiation.....	14,400	8,150
q_2	b. Based on condensate corrected for radiation.....	8,320	2,070
q_3	c. Based on air temperatures.....	12,350	6,330
	<i>Overall coefficients, B.t.u./(hr. \times ft.² \times °F.)</i>		
U_1	a. Based on uncorrected condensate.....	4.50	2.70
U_2	b. Based on corrected condensate.....	2.60	0.70
U_3	c. Based on air temperatures.....	3.86	2.09
U_4	d. Based on film-coefficient equation.....	4.24	3.18

Calculations for Run No. 1**1. Area of tempering coil, A^* , ft.²**

$$\begin{aligned}
 &= \pi \frac{(\text{O.D. copper tubes, in.})}{12} \times \frac{(\text{length tubes, in.})}{12} \times \text{No. of tubes} \\
 &= \pi \left(\frac{1.125}{12} \right) \times \frac{18}{12} \times 45 = 19.90
 \end{aligned}$$

* In these calculations, symbols refer to quantities in Tables 2 and 3.

2. *Logarithmic-mean temperature difference, Δt , °F.*

$$= \frac{(t_1 - t_2) - (t_1 - t_3)}{2.303 \log_{10} \frac{(t_1 - t_2)}{(t_1 - t_3)}} = \frac{(264 - 74.2) - (264 - 127.6)}{2.303 \log_{10} \frac{(264 - 74.2)}{(264 - 127.6)}}$$

$$= 161$$

3. *Density of exit air, ρ , lb./ft.³*

$$= \frac{\text{Mol. weight air}}{359} \times \frac{(460 + 32)}{(460 + t_3)} \times \frac{P_1}{29.92}$$

$$= \frac{29}{359} \times \frac{492}{587.6} \times \frac{30.19}{29.92} = 0.0683$$

4. *Quantity of air, G , lb./hr.*

$$= \rho \times V \times 60 \times (\text{cross-sectional area of duct, ft.}^2)$$

$$= 0.0683 \times 674.2 \times 60 \times [(8/12)^2 \times (\pi/4)]$$

$$= 0.0683 \times 674.2 \times 60 \times 0.349 = 964$$

5. *Heat transferred, B.t.u./hr.*a. q_1 , based on uncorrected condensate

$$= W_1 \times (\text{latent heat of steam at } t_1) = 15.4 \times 936 = 14,400$$

b. q_2 , based on corrected condensate

$$= (W_1 - W_0) \times (\text{latent heat of steam at } t_1) = (15.4 - 6.50) \times 936$$

$$= 8,320$$

c. q_3 , based on air temperatures

$$= (\text{av. sp. ht. of the air}) \times (t_4 - t_2) \times G$$

$$= 0.24 \times (127.6 - 74.2) \times 964 = 12,350$$

6. *Experimental Overall Heat-Transfer Coefficients, U .*

$$U = \frac{q}{A(\Delta t)} \quad (1)$$

where

U = overall coefficient, B.t.u./hr. \times ft.² \times °F.).

q = heat transferred, B.t.u./hr.

A = area of heating surface, ft.²

Δt = log-mean temp. difference, °F.

a. U_1 , based on uncorrected condensate

$$= \frac{q_1}{A(\Delta t)} = \frac{14,400}{19.90 \times 161} = 4.50$$

b. U_2 , based on corrected condensate

$$= \frac{q_2}{A(\Delta t)} = \frac{8,320}{19.90 \times 161} = 2.60$$

c. U_1 , based on air temperatures

$$= \frac{q_1}{A(\Delta t)} = \frac{12,350}{19.90 \times 161} = 3.86$$

7. *Calculation of the air-film coefficient, h_a .*

$$\begin{aligned} h_a &= 0.131 \left(\frac{k_f}{D} \right) \left(\frac{Dv_{\max}}{\mu_f} \right)^{0.7} \\ &= 0.131 \times \frac{0.0159}{0.0937} \times \left(\frac{0.0937 \times 964}{2.42 \times 0.0205} \right)^{0.7} \\ &= 0.131 \times 0.1696 \times 191.4 = 4.26 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}). \end{aligned} \quad (2)$$

where

$k_f = 0.0159 \text{ B.t.u.-ft.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}) = \text{thermal conductivity of the air film.}$

$\mu_f = (2.42 \times 0.0205) \text{ lb.}/(\text{hr.} \times \text{ft.}) = \text{viscosity of the air film.}$

$D = \frac{1.125}{12} = 0.0937 \text{ ft.} = \text{diameter of the tubes.}$

$v_{\max} = 964 \text{ lb.}/(\text{hr.} \times \text{ft.}^2) = \text{mass velocity.}$ (The quantity of air was 964 lb./hr. and the free area of the first row of eight tubes is 1 ft.²).

The values of k_f and μ_f are based on a film temperature of 183°F., which is the mean of 264°F.—the steam temperature—and 101°F.—the average air temperature.

8. *Overall Heat-Transfer Coefficient, U_1 , Based on Film Coefficients.*

$$\begin{aligned} U_1 &= \frac{1}{\frac{1}{h_a} + \frac{d_1 L}{d_2 k} + \frac{d_1}{d_2 h_s}} \\ &= \frac{1}{\frac{1}{4.26} + \frac{1.125 \times 0.00521}{1.0625 \times 222} + \frac{1.125}{1 \times 1500}} \\ &= \frac{1}{0.235 + 0.000025 + 0.00075} = \frac{1}{0.236} \\ &= 4.24 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}). \end{aligned} \quad (3)$$

where

$h_a = \text{air-film coefficient} = 4.26 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}).$

$k = \text{thermal conductivity of copper}$

$= 222 \text{ B.t.u.-ft.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}).$

h_s = steam-film coefficient = 1500 B.t.u./ (hr. \times ft.² \times °F.)
[assumed value].

d_1 = O.D. of tubes = 1.125 in.

d_2 = I.D. of tubes = 1.0 in.

$d_{av} = \frac{1}{2}(d_1 + d_2) = 1.0625$ in.

L = thickness of the copper tubes, ft.

$= \frac{1}{2}(d_1 - d_2) \times 1/12 = 0.0625/12 = 0.00521$

Comments. 1. It is evident from Table 3, that the overall coefficients based on corrected condensate do not agree with those based on air temperatures, although the latter agree more closely with those calculated from the film-coefficient equation. A considerable part of the discrepancy may have been due to inaccuracies in the weights of condensate, because of the relatively small amount involved, and because the steam trap did not allow a steady flow of liquid.

2. Radiation losses were very large because the drier was not insulated. This is not particularly serious if a piece of equipment is used only occasionally, but if it is used extensively, it should be insulated.

NOMENCLATURE

A = area, ft.²

d = diameter, in.

D = diameter, ft.

G = flow rate, lb./hr.

h = film coefficient, B.t.u./ (hr. \times ft.² \times °F.).

k = thermal conductivity, B.t.u.-ft./ (hr. \times ft.² \times °F.).

L = thickness, ft.

P_1 = air pressure, in. of Hg.

P_2 = steam pressure, lb./in.² gage.

q = heat transferred, B.t.u./hr.

t = temperature, °F.

Δt = temperature difference, °F.

U = overall coefficient, B.t.u./ (hr. \times ft.² \times °F.).

v_{max} = maximum mass velocity, lb./ (hr. \times ft.²).

V = velocity, ft./min.

W = steam condensed, lb./hr.

μ = viscosity, lb./ (hr. \times ft.).

ρ = density, lb./ft.³

REFERENCE

1. BADGER AND McCABE, "Elements of Chemical Engineering," McGraw-Hill Book Company, New York, 1936, 2nd ed., p. 137.

A NITRATOR

Designed and Constructed by
The School of Chemical Engineering
Cornell University*

Description. This nitrator is constructed on somewhat the same principle as the Hough nitrator.¹ It consists of two parts: a vertical, cylindrical reaction chamber that is provided with a cooling coil, and a circulation chamber—a vertical cylinder fitted with propellers for circulating the charge. The reaction chamber is connected to the circulation chamber by means of flanges so that reaction chambers of different sizes can be used with the same circulation chamber. The equipment is made largely from standard steel pipe, and all joints and flanges are welded. The nitrator described here is provided with two reaction chambers: one made from 8-inch pipe and the other from 12-inch pipe. Figure 1 is a photograph of the complete assembly with the 12-inch chamber; and the dimensions of both chambers are given in Figure 2.

Materials and Costs. The materials used in constructing the nitrator assembly with both the 12-inch and 8-inch reaction chambers are given in Table 1. The approximate costs of these materials are:

1. For the 12-inch reaction chamber: \$25.
2. For the 8-inch reaction chamber: \$20.
3. For the circulation chamber assembly: \$25.
4. For supports and insulation: \$7.

Notes on Construction. 1. The reaction chambers are provided with welded dished-bottoms and flanged top plates. Flanged sections of 2-inch pipe are welded to openings in the side and bottom of each chamber, as shown in Figure 1.

* Reported by C. C. Winding and F. H. Rhodes.

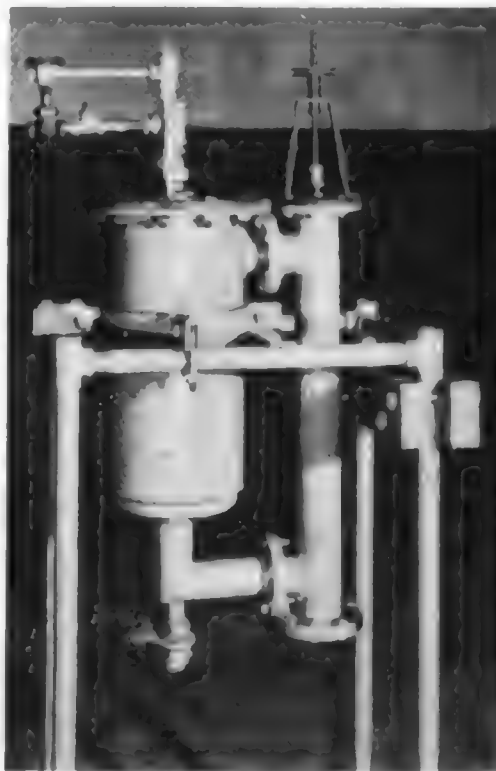


Figure 1. Laboratory Nitrator—Cornell University

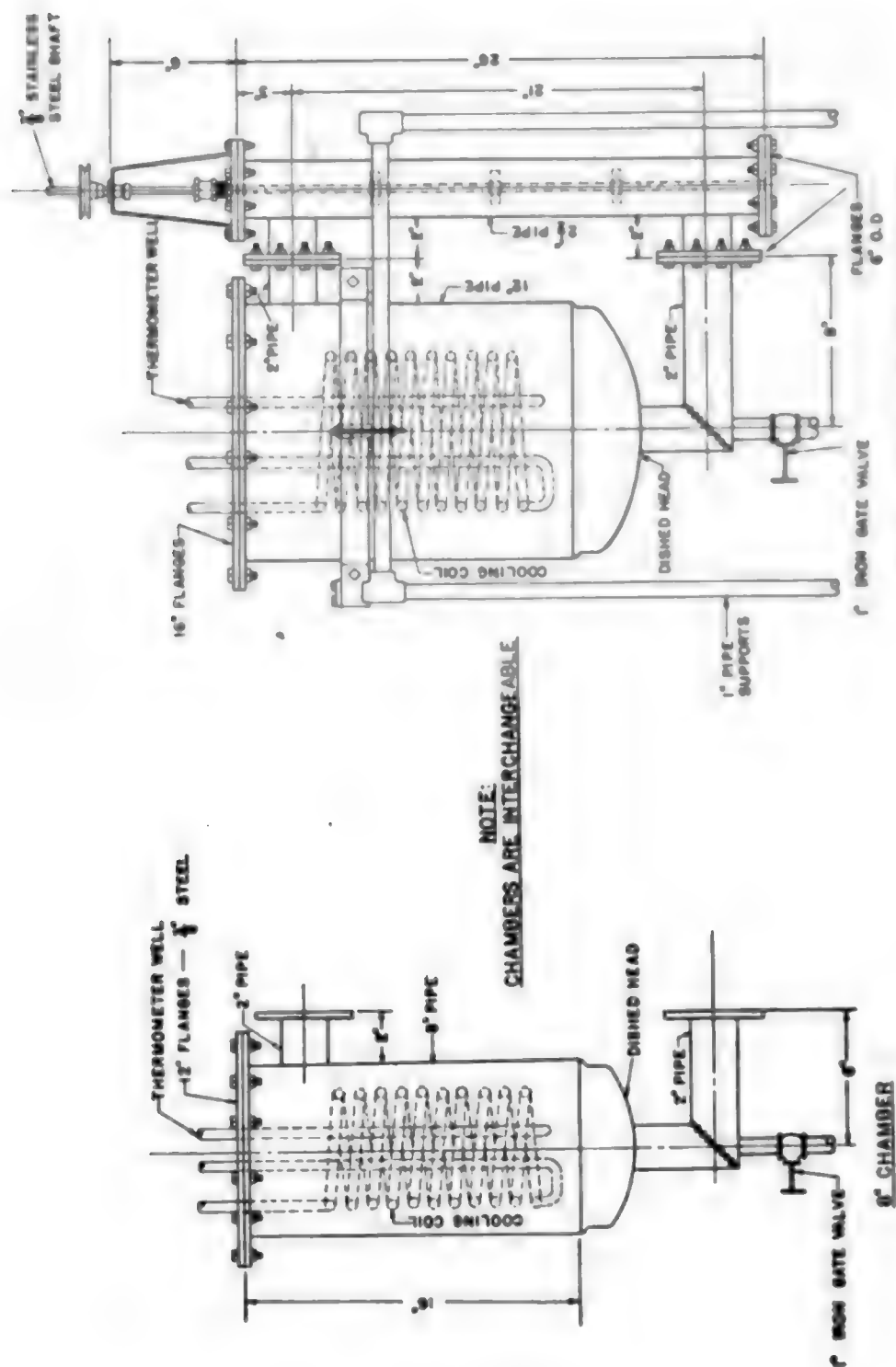


TABLE 1
Bill of Materials for Nitrator
 Cornell University

Quantity	Description	Size	Material
For 12-inch reaction chamber			
16 in.	Std. pipe	12 in.	Black wrought steel
19 in.	" "	2 in.	" " "
10 ft.	Pipe	0.31 in. I.D. x $\frac{1}{2}$ in. O.D.	Steel
1	Dished head	12 in. O.D.	"
1	Flange	12 in. I.D. x 16 in. O.D. x $\frac{1}{2}$ in. thick	"
2	"	2 in. I.D. x 6 in. O.D. x $\frac{1}{2}$ in. thick	"
1	Round plate	16 in. D. x $\frac{1}{2}$ in. thick	"
1	Gate valve	1 in.	All iron
For 8-inch reaction chamber			
16 in.	Std. pipe	8 in.	Black wrought steel
14 in.	" "	2 in.	" " "
10 ft.	Pipe	0.31 in. I.D. x $\frac{1}{2}$ in. O.D.	Steel
1	Dished head	8 inch	"
1	Flange	8 in. I.D. x 12 in. O.D. x $\frac{1}{2}$ in. thick	"
2	"	2 in. I.D. x 6 in. O.D. x $\frac{1}{2}$ in. thick	"
1	Round plate	12 in. D. x $\frac{1}{2}$ in. thick	"
1	Gate valve	1 in.	All iron
For circulation chamber assembly			
26 in.	Std. pipe	2 $\frac{1}{2}$ in.	Black wrought steel
2	Flange	2 $\frac{1}{2}$ in. I.D. x 6 in. O.D. x $\frac{1}{2}$ in. thick	Steel
2	"	2 in. I.D. x 6 in. O.D. x $\frac{1}{2}$ in. thick	"
2	Round plate	6 in. D. x $\frac{1}{2}$ in. thick	"
3 ft.	Shafting	$\frac{1}{2}$ in.	Stainless steel
3	Impellers	2 in.	High-nickel cast steel
1	Bearing assembly and support	$\frac{1}{2}$ in.	Brass and steel
1	Stuffing box	$\frac{1}{2}$ in.	Brass
1	Pulley	$\frac{1}{2}$ in. shaft	Cast iron
1	Electric motor	$\frac{1}{2}$ H.P.	—

TABLE 1—*Concluded*

Quantity	Description	Size	Material
For supports			
19½ ft.	Std. pipe	1 in.	Black wrought steel
4	Side-outlet railing tee	1 in.	Black malleable iron
4	Floor flange	1 in.	" " "
7 ft.	Flats	¼ x 1½ in.	Steel •
6 ft.	Lumber	2 x 4 in.	Pine
Miscellaneous items			
1 doz.	Bolts and nuts	¾ in.—16 NC x 1½ in. long	Steel
2 doz.	" " "	¾ in.—20 NC x 1½ in. long	"
2 ft.	Pipe insulation	2½ in.	85% magnesia
2 ft.	" "	2 in.	" "
16 in.	" "	8 in.	" "
16 in.	" "	12 in.	" "

2. An all-iron 1-inch gate valve, attached to a nipple welded into the line at the bottom of the reaction chamber, serves as a discharge port.

3. The cooling coil is made of steel pipe, ¾-inch O.D. x 0.31-inch I.D. The leads to and from the coil pass through openings in the cover plate, and are held in place by packing glands and lock nuts. The required length of coil depends upon the amount of heat liberated in the particular nitration being carried out, but a coil of 5 turns, each 5½ inches in diameter, is sufficient for most nitrations in the 8-inch chamber. A coil of these dimensions has a total effective length of 112 inches. The coils illustrated in Figure 1 are considerably larger than this.

4. A valve is installed in the inlet line to control the flow of cooling water through the coil, and the rate of flow is measured by an orifice weir in the outlet line. The coil is also connected to a steam line so that it may be used for heating as well as for cooling.

5. Thermometer wells are provided in both the inlet and outlet lines of the coil, and another thermometer well, 10 inches deep, extends through the cover plate of the reaction chamber.

6. A hole of sufficient size to take the stem of a large glass separatory funnel is drilled in the cover plate to provide a charging opening.

7. The circulation chamber is made of a section of 2½-inch pipe, provided with a central shaft carrying three upward-thrusting pro-

pellers; the pipe has two flanged side openings for connection to the reaction chamber. The shaft is suspended from a support carrying thrust bearings. Excessive *whipping* is prevented by a loose collar bearing that rests on the bottom end plate.

8. The entire apparatus is supported on a framework made of standard 1-inch pipe and railing fittings.

9. All parts of the nitrator assembly, with the exception of the cover plate, are insulated with 85 per cent magnesia.

Operation. Although many different nitrations can be carried out in this apparatus, the nitration of benzene is typical. In the commercial manufacture of mononitrobenzene, benzene is nitrated with a *mixed acid*, which is a mixture of sulfuric and nitric acids of approximately the following percentage composition: 63 per cent H_2SO_4 , 36 per cent HNO_3 , and 1 per cent H_2O .

The sulfuric acid serves two purposes: (1) it inhibits the oxidizing action of the nitric acid, and (2) it absorbs the water formed by the reaction



and thereby permits the nitration to proceed to substantial completion.

To prevent sudden changes in concentration and in the temperature of the mixture, it is customary to effect the reaction in the presence of some *spent acid* from a previous nitration. Such spent acid is usually referred to as *cycle acid*.

The following procedure is typical of commercial operations:

1. The cycle acid is first delivered to the nitrator and cooled to 50°C . (122°F .) or lower, the benzene is then added, and finally the mixed acid is run in slowly either on top of the benzene or beneath its surface. The temperature in the nitrator is kept below 60°C . (140°F .) while the mixed acid is being added.

2. After nitration is complete, the charge is run into conical-bottom lead tanks and settled for approximately 12 hours. The lower layer of spent acid, containing some nitrobenzene, is drawn off and sent to spent-acid tanks for additional settling or for extraction with benzene to remove residual nitrobenzene; this benzene is subsequently used in a nitration charge. About one-third of the spent acid from a nitration is added as cycle acid to a subsequent charge and the remainder is treated for the recovery of the acids.²

In carrying out nitrations in the laboratory nitrator, it is necessary to depart from the usual commercial procedure in three respects: (1) to provide effective agitation at the beginning of the experiment, it is

necessary to use a somewhat larger quantity of cycle acid; (2) the cycle acid is not the spent acid from a previous charge but is prepared by diluting ordinary concentrated sulfuric acid; and (3) the mixed acid is prepared from ordinary concentrated sulfuric and nitric acids and, therefore, contains more water than the mixed acid used commercially.

TYPICAL EXPERIMENT

Object. To determine the rate of heat transfer through the cooling coil of a nitrator during the nitration of benzene.

Procedure. 1. Prepare a quantity of cycle acid in the following manner:

- a. Introduce about 40 pounds of 93 per cent sulfuric acid into the dry nitrator.
- b. Pass cooling water through the coil.
- c. Start the agitator.
- d. Slowly add sufficient water to give a final concentration of 73 per cent H_2SO_4 .
- e. Cool the mixture and run it into carboys for later use.

2. In a similar manner, prepare about 20 pounds of mixed acid from 13.1 pounds of 93 per cent sulfuric acid and 6.9 pounds of 67 per cent nitric acid. Store it in carboys for subsequent use.

3. Pour 30 pounds of the cycle acid into the nitrator and then add 5.1 pounds of benzene. Start the agitator, and pass steam through the coil. When the temperature of the charge reaches 54°C . (130°F .), shut off the supply of steam and pass cold water through the coil at a rate of about 1 gallon per minute.

4. When the temperature of the charge has dropped to 50°C . (122°F .), begin adding the mixed acid through a heavy glass separatory funnel inserted in the charging opening of the nitrator. The rate of addition of the acid should be such that the temperature of the charge remains at about 49°C . (120°F .).

5. At the instant the addition of mixed acid is begun—the zero time for the experiment—divert the outlet of the stream of cooling water into a tared drum.

6. Continue the addition of mixed acid until a total of 17.8 pounds have been added. At the instant when the last of the mixed acid has been added, divert the cooling water to waste and weigh the quantity of cooling water collected.

7. Record the temperature of the mixed acid at the beginning of the

TABLE 2

Experimental Data and Calculated Results for Heat Transfer in a Nitrator

Symbol	Coil Measurements			
d_i	Inside diameter, in.....	0.31		
d_o	Outside diameter, in.....	0.625		
n	Total effective length, in.....	112		
	Item	Run No.		
		1	2	3
θ	Timed interval, min.....	25	27	22.5
t_i	Mean inlet temp. of water, °C.....	9.2	11.2	11.0
t_o	Mean outlet temp. of water, °C.....	19.8	28.1	31.3
t_a	Av. temp. of water, °C.....	14.5	19.6	21.1
t_a	Av. temp. of water, °F.....	58.1	67.4	70.0
t_s	Rise in temp. of water, °C.....	10.6	16.9	20.3
t_s	Rise in temp. of water, °F.....	19.1	30.5	36.5
W	Weight of water, lb.....	196	139	104
k_w	Av. thermal conductivity of water, B.t.u.- ft./ (hr. x ft. ² x °F.).....	0.339	0.344	0.346
μ	Av. viscosity of water, lb./ (ft. x hr.).....	2.8	2.45	2.37
t_c	Mean temp. of charge, °F.....	118.4	123.6	136.5
Δt	Mean temp. difference, charge to water, °F.....	60.3	56.2	66.5
V	Velocity of water, ft./hr.....	13,850	8,790	7,290
U	Overall coefficient, B.t.u./ (hr. x ft. ² x °F.).....	130.6	141.5	126.8
h_i	Inner-film coefficient, B.t.u./ (hr. x ft. ² x °F.).....	900	670.5	626
H_i	Inner-film conductance, B.t.u./ (hr. x °F. x linear ft.).....	73.0	54.5	50.4
H_w	Thermal conductance of wall, B.t.u./ (hr. x °F. x linear ft.).....	234.5	234.5	234.5
H	Overall conductance, B.t.u./ (hr. x °F. x linear ft.).....	15.45	16.73	15.0
H_o	Outer-film conductance, B.t.u./ (hr. x °F. x linear ft.).....	21.4	27.0	23.45
h_o	Outer-film coefficient, B.t.u./ (hr. x ft. ² x °F.).....	130.0	165.0	142.3

run, and the following readings at zero time and one minute intervals thereafter until the last of the acid has been added:

- Temperature of the charge.
- Temperature of the inlet cooling water.
- Temperature of the outlet cooling water.
- Reading of the liquid level in the weir box in the outlet water line.

8. After all of the mixed acid has been added, continue the agitation until there is no further rise in temperature, but in no case discontinue agitation until at least 15 minutes have elapsed since the addition of the last of the mixed acid.

9. Cool the charge to about 27°C. (80°F.), and draw it off into a carboy.

10. If the nitrator is not to be reused immediately, wash it out thoroughly with a solution of sodium carbonate.

11. Calculate:

- a. The overall heat-transfer coefficient from the data obtained in the experiment.

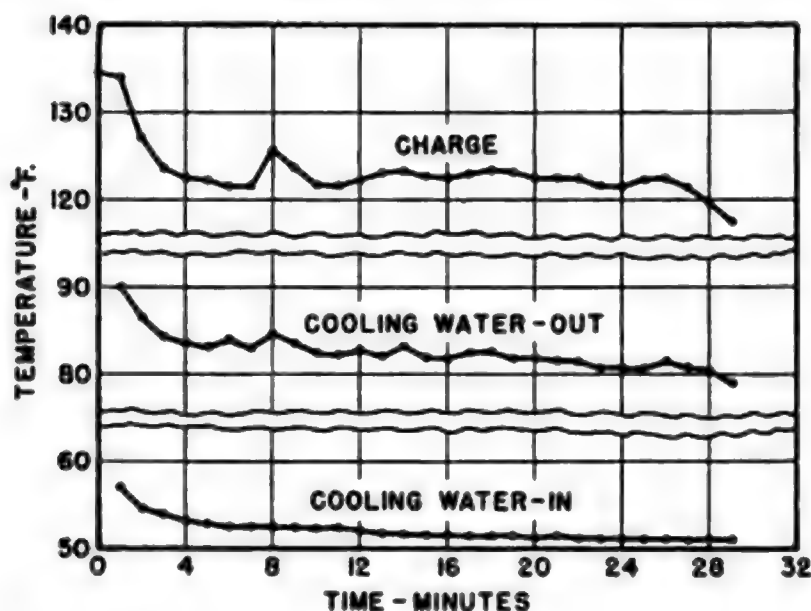


Figure 3. Temperature Log for Run No. 2 of Nitration Experiment

- b. The inner-film coefficient by means of the *Dittus-Boelter* equation.
- c. The outer-film coefficient from the overall coefficient and the inner-film coefficient.

Typical Data. The experimental data for three runs are given in Table 2, and a log of the temperatures for Run No. 2 is shown in Figure 3.

CALCULATIONS FOR RUN NO. 1

Cooling Coil.

$$D_i = \text{inner diameter, ft.} = \frac{d_i}{12} = 0.31/12 = 0.02585$$

$$D_o = \text{outer diameter, ft.} = \frac{d_o}{12} = 0.625/12 = 0.05208$$

$$D_m = \text{mean diameter, ft.}$$

$$= \frac{(0.05208 - 0.02585)}{2.303 \log_{10} \frac{0.05208}{0.02585}} = 0.03739$$

$$L = \text{wall thickness, ft.} = \frac{1}{2}(0.05208 - 0.02585) = 0.01312$$

$$N = \text{length, ft.} = \frac{n}{12} = 112/12 = 9.33$$

$$A_i = \text{total inner area, ft.}^2 \\ = \pi \times D_i \times N = 3.1416 \times 0.02585 \times 9.33 = 0.758$$

$$A_o = \text{total outer area, ft.}^2 \\ = \pi \times D_o \times N = 3.1416 \times 0.05208 \times 9.33 = 1.53$$

$$A_m = \text{total mean area, ft.}^2 \\ = \pi \times D_m \times N = 3.1416 \times 0.03739 \times 9.33 = 1.10$$

$$a = \text{inner cross-sectional area, ft.}^2 \\ = \pi \times \frac{D_i^2}{4} = 3.1416 \times \frac{(0.02585)^2}{4} = 0.000525$$

$$k_s = \text{thermal conductivity of steel, B.t.u.-ft.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}) \\ = 26.2$$

Temperatures.

$$t_1 = \text{av. temp. of water, } ^\circ\text{F.} = 1.8 (t_2) = 1.8 (t_1 + t_2)/2 \\ = 1.8 (9.2 + 19.8)/2 = 58.1$$

$$t_2 = \text{rise in temp. of water, } ^\circ\text{F.} = 1.8 (t_3) = 1.8 (t_2 - t_1) \\ = 1.8 (19.8 - 9.2) = 19.1$$

$$\Delta t = \text{temp. difference, charge to water, } ^\circ\text{F.} = t_2 - t_1 \\ = 118.4 - 58.1 = 60.3$$

Physical data of water at 58.1°F.

$$\mu = \text{viscosity, lb.}/(\text{ft.} \times \text{hr.}) = \text{viscosity in centipoises} \times 2.42 \\ = 1.157 \times 2.42 = 2.8$$

$$k_w = \text{thermal conductivity, B.t.u.-ft.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}) = 0.339$$

$$\rho = \text{density, lb.}/\text{ft.}^3 = 62.4$$

$$V = \text{velocity, ft.}/\text{hr.}$$

$$= W \times 60/\theta \times 1/\rho \times 1/a$$

$$= 196 \times 60/26 \times 1/62.4 \times 1/0.000525 = 13,850$$

$$c_p = \text{heat capacity, B.t.u.}/(\text{lb.} \times ^\circ\text{F.}) = 1.0$$

Coefficients and Conductances.

$$U = W \times 60/\theta \times t_2 \times c_p \times 1/A_m \times 1/\Delta t \\ = 196 \times 60/26 \times 19.1 \times 1.0 \times 1/1.10 \times 1/60.3 \\ = 130.6$$

h_i = inner-film coefficient, B.t.u./ $(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.})$ (from Dittus-Boelter equation)

$$\begin{aligned}
 &= 0.0225 \frac{k_w}{D_i} (D_i V \rho / \mu)^{0.8} (c_p \mu / k_w)^{0.4} \\
 &= 0.0225 \left(\frac{0.339}{0.02585} \right) \left(\frac{0.02585 \times 13,850 \times 62.4}{2.8} \right)^{0.8} \left(\frac{1.0 \times 2.8}{0.339} \right)^{0.4} \\
 &= 900
 \end{aligned}$$

H_i = inner-film conductance per linear foot of pipe, B.t.u./ $(\text{hr.} \times ^\circ\text{F.} \times \text{ft.})$

$$\begin{aligned}
 &= h_i \times \pi D_i = 900 \times 3.1416 \times 0.02585 \\
 &= 73.0
 \end{aligned}$$

H_s = thermal conductance of wall per linear foot of pipe, B.t.u./ $(\text{hr.} \times ^\circ\text{F.} \times \text{ft.})$

$$\begin{aligned}
 &= k_s \times \pi D_m / L = 26.2 \times \frac{3.1416 \times 0.03739}{0.01312} \\
 &= 234.5
 \end{aligned}$$

H = overall conductance per linear foot, B.t.u./ $(\text{hr.} \times ^\circ\text{F.} \times \text{ft.})$

$$\begin{aligned}
 &= W \times t_b \times c_p \times 60 / \theta \times 1 / \Delta t \times 1 / N \\
 &= 196 \times 19.1 \times 1.0 \times 60 / 26 \times 1 / 60.3 \times 1 / 9.33 \\
 &= 15.45
 \end{aligned}$$

$$1/H = 1/H_i + 1/H_s + 1/H_o$$

Therefore,

$$\begin{aligned}
 1/H_o &= 1/H - 1/H_i - 1/H_s \\
 &= 1/15.45 - 1/73.0 - 1/234.5 \\
 &= 0.0647 - 0.01370 - 0.00426 \\
 &= 0.0467
 \end{aligned}$$

and

$$H_o = 1/0.0467 = 21.4$$

h_o = outer-film coefficient, B.t.u./ $(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.})$

$$\begin{aligned}
 &= H_o / (\pi \times D_o) = 21.4 / (3.1416 \times 0.05208) \\
 &= 130.0
 \end{aligned}$$

NOMENCLATURE

a = cross-sectional area, ft.^2

A = surface area, ft.^2

c_p = heat capacity, B.t.u./ $(\text{lb.} \times ^\circ\text{F.})$.

- d = diameter, in.
 D = diameter, ft.
 h = film coefficient of heat transfer, B.t.u./($\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}$).
 H = heat conductance, B.t.u./($\text{hr.} \times ^\circ\text{F.} \times \text{linear ft.}$).
 k = thermal conductivity, B.t.u.-ft./($\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}$).
 L = wall thickness, ft.
 n = length of cooling coil, in.
 N = length of cooling coil, ft.
 t = temperature, $^\circ\text{F.}$
 Δt = temperature difference, $^\circ\text{F.}$
 U = overall heat transfer coefficient, B.t.u./($\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}$).
 V = velocity, ft./hr.
 θ = time of run, min.
 μ = viscosity, lb./($\text{ft.} \times \text{hr.}$).
 ρ = density, lb./ft.³

Subscripts

- i = inner
 m = mean
 o = outer
 s = wall
 w = water

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1. Groggins, "Unit Processes in Organic Synthesis," McGraw-Hill Book Company, New York, 2nd ed., 1938, p. 30.
2. Ibid., pp. 33-40.

AN ELECTRIC MUFFLE FURNACE

Designed and Constructed by
The Engineering Experiment Station
University of New Hampshire*

Description. This is an electric muffle furnace of the resistance type, having internal dimensions of 9 inches in width x 9 inches in depth x 10 inches in height. The heating elements are made of a special alloy resistance wire which is capable of withstanding furnace temperatures as high as 2300°F. The furnace is operated directly from a 110-120 volt A.C. circuit, and draws only about 25 amperes. One feature of this furnace is that the muffle is made from insulating firebrick, thus eliminating the need for a commercial type of muffle.

A front and a rear view of the assembled furnace are shown in Figure 1, and details of construction are shown in Figures 2 and 3.

Materials and Cost. Approximately \$50 worth of materials, as listed in Table 1, are required for the construction of this furnace; the total time required for its construction is about 50 man-hours.

Notes on Construction. The door and muffle (Figure 2) are made from 9-inch straight insulating firebrick, which are grooved to receive the heating elements. These bricks are covered on the outside with a 2-inch layer of insulation blocks, and the whole structure is then enclosed in a shell of $\frac{1}{4}$ -inch asbestos sheathing, cut as shown in Detail A of Figure 3, and held together at the edges with steel angles.

Grooving of the Bricks. The grooves for the heating elements are cut into the bricks as shown in Figure 2. The cuts are made by holding the brick against a guide on a drill press table and pushing it against a specially constructed tool, which is made by cutting the tip from a $\frac{1}{4}$ -inch wood bit and replacing it with a cutter $1\frac{1}{4}$ -inch in diameter and either $\frac{5}{32}$ -inch or $\frac{3}{16}$ -inch thick. The cutter should rotate about 200 R.P.M.

As an added precaution, the surfaces of the grooves which come in contact with the resistance wires may be coated with an inert refractory cement.† This is not absolutely essential, for the bricks are usually free from impurities which react with the wire; however, it does give greater insurance against resistor wire burnout. In cases where the grooves are not coated and wire failure does occur, it is essential that the slagged spots be coated with cement before the wire is replaced.

* Reported by L. E. Webber and D. S. Eppelsheimer.

† Such as No. RA-1161, Norton Company, Worcester, Mass.

Assembly of Muffle and Door. The muffle is most easily assembled on a sturdy bench or table about 30 inches high. Six or seven common red bricks, or other blocking about 2 inches high, are first placed on the table, and on top of these is laid the piece of sheathing which has been cut for the bottom. The blocking should be an inch or two inside the

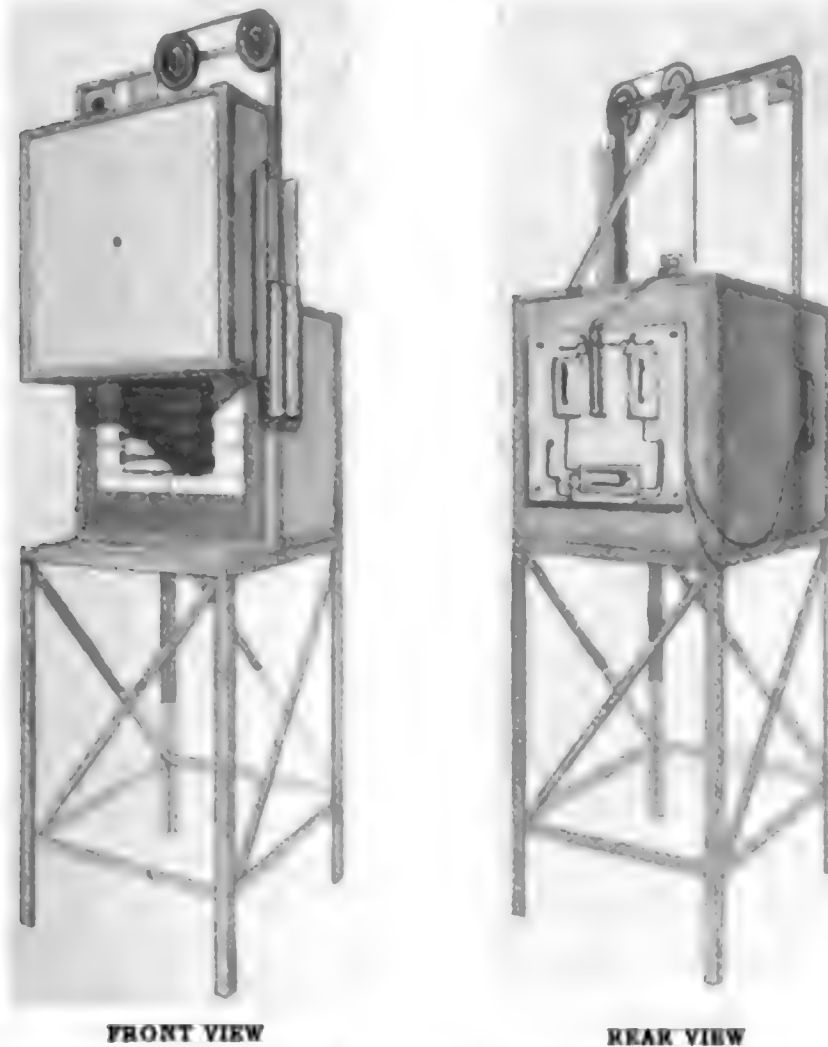


Figure 1. Electric Muffle Furnace—University of New Hampshire

edge of the sheathing so as not to interfere with the placement of the steel frame later in the assembly.

The insulation blocks for the bottom of the furnace are laid on top of the sheathing, and the bricks for the muffle are assembled as shown; the bricks are covered with the insulation blocks; and finally the asbestos sheets for the sides, back, and top are put in place and secured by the steel frame. During construction, it is advisable to support the top

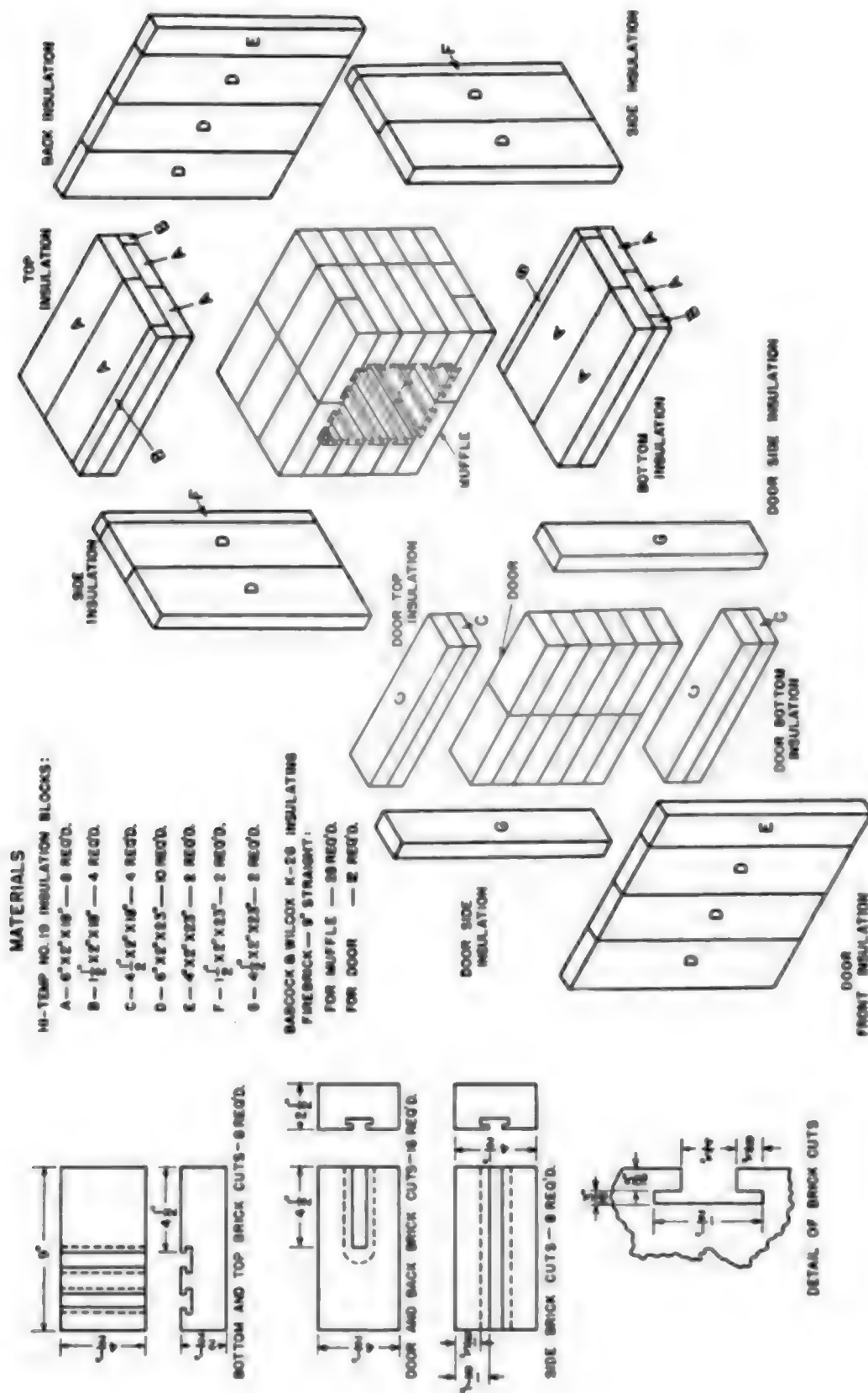


Figure 2. Details of Construction of Door and Muffle

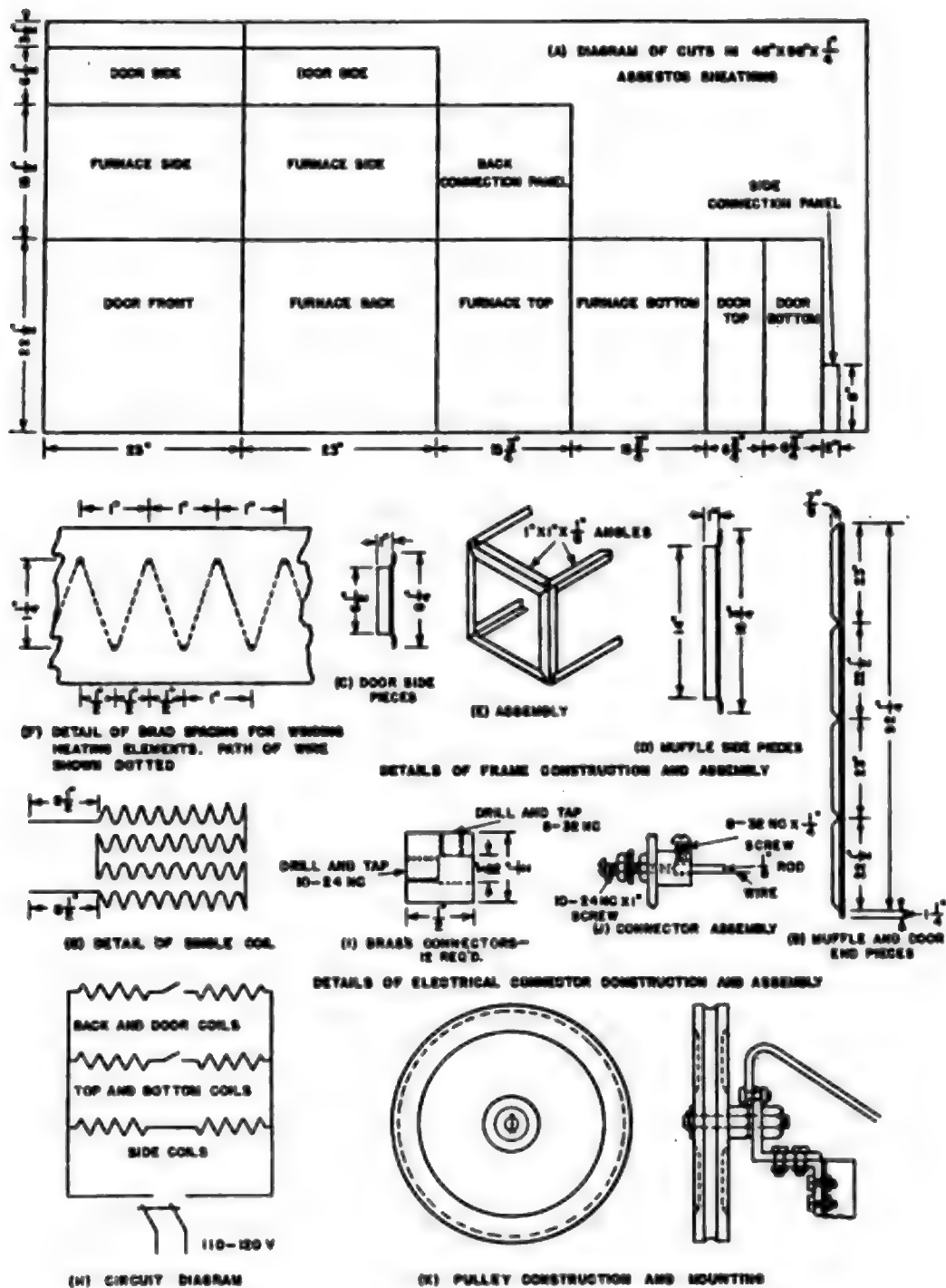


Figure 3. Details of Construction of Muffle Furnace

with a small jack. After the muffle is assembled, a $\frac{1}{4}$ -inch hole is drilled through the center of the back to receive a chromel-alumel thermocouple made of No. 8 B. & S. gage wire inserted through 2-hole porcelain insulators.

TABLE 1
Bill of Materials for Electric Muffle Furnace
University of New Hampshire

Quantity	Description	Size	Material
40	Insulating firebrick	9 in. straight	Babcock and Wilcox, K-26*
15	Insulation blocks	6 x 2 x 36 in.	Hi-Temp. No. 19**
0.8 lb.	Resistance wire ribbon	3 x 3 mm.	Kanthal A-1***
1 piece	Asbestos sheathing	48 x 96 x $\frac{1}{4}$ in.	Careystone**
44 ft.	Angles	1 x 1 x $\frac{1}{4}$ in.	Steel
2	Pulley	6 in. diam.	"
2	Weight	—	Cast iron
2	Electrical switch (Double-pole, single-throw)	—	Copper
1	Electrical switch (Single-pole, single-throw)	—	"
3 ft.	Flats	$\frac{1}{2}$ x $\frac{1}{4}$ in.	Steel
4 ft.	Wire rope	$\frac{1}{2}$ in.	"
3 ft.	Electrical wire	No. 10 gage	Copper
4 ft.	BX cable	No. 14 gage	"
2 doz.	Cap screws	$\frac{1}{2}$ in.-20 NC x $\frac{1}{4}$ in. long	Steel
8	Wood screws	No. 10 x 2 in. long	"
$\frac{1}{2}$ doz.	Bolts and nuts	$\frac{1}{2}$ in.-20 NC x $\frac{1}{4}$ in. long	"
1 doz.	Connectors	—	Brass
2 ft.	Thermocouple wire	No. 8 gage	Chromel-Alumel
2 doz.	Thermocouple insulators (2 hole)	For No. 8 wire	Porcelain
2	Window fasteners	—	Steel

* Babcock and Wilcox Co., 85 Liberty St., New York.

** Philip Carey Mfg. Co., 267 Medford Street, Boston, Mass.

*** C. O. Jelliff Mfg. Corp., Southport, Connecticut.

In assembling the door, the sheathing for the front is first laid on the blocking, and then the insulation blocks and bricks are put in place.

After the door is assembled, a spy hole, about an inch in diameter, is drilled in it. A suitable plug for this hole can be filed from a small piece of firebrick.

Frames. The frames for both the muffle and the door are made in the same manner. The front and back sections of each frame are bent from two pieces of angles, cut as shown in Detail *B* of Figure 3, and the final corner of each frame is secured with two small screws. These sections are then slipped over the sheathing, and the four edge pieces, cut and forged as shown in Details *C* and *D*, are attached at each corner with $\frac{1}{4}$ inch-20NC \times $\frac{1}{4}$ inch long cap screws, threaded into the frame (Detail *E*).

Heating Elements. The heating elements consist of six coils of 3×0.3 mm. Kanthal wire, each 14.6 feet long, wound as shown in Details *F* and *G* of Figure 3, and hooked-up in a combination parallel-series arrangement, as illustrated in Detail *H*. About $8\frac{1}{2}$ inches of wire at the ends of each coil are left unformed for passage through the insulation of the connector panel. The ends of the wires entering the connectors are later bent double. It is also good practice, although not essential, to make the entire ends of the wires two or three times their original thicknesses where they pass through the insulation. This can be done by spot welding one or two additional pieces of wire along the ends of the coils. When this is done, the coils should be made slightly longer to compensate for the decreased resistance of the ends.

In winding the coils, the wire is marked off $8\frac{1}{2}$ inches from each end, and the length of the wire between the marks is laid off into four equal divisions. Each marked section is formed individually by winding the wire around brads driven in a board according to Detail *F*, leaving enough unformed wire between the individual sections to give sufficient slack for ease in assembly.

Each coil wound in this manner will fill four grooves in the muffle. If desired, individual coils of sufficient length for only two grooves can be used; this makes it easier to replace burned out elements, but it requires a greater number of connections outside the furnace.

The length of wire per coil was determined by the following computations:

1. The furnace was designed to operate at 25 amperes and 115 volts; therefore, each of the three parallel sets of coils would have to carry $25/3$ or 8.33 amperes.
2. As $R = E/I$, the resistance of each set of coils must be $115/8.33$ or 13.85 ohms.
3. Kanthal wire, 3×0.3 mm., has a resistance of 0.475 ohms/ft.; therefore, the length of wire for each parallel set of coils is $13.85/0.475$ or 29.2 feet.
4. As there are two coils in series in each parallel set, the length of wire per coil is $29.2/2$ or 14.6 feet.

The maximum permissible furnace temperature can be computed from the area of the wire and the energy consumption. The wire has an area of approximately 3 square inches per foot, and each set of parallel coils—29.2 feet of wire—must carry 8.33 amperes. Therefore, the energy concentration is $(115 \times 8.33)/(3 \times 29.2) = 11$ watts per square inch. From the manufacturer's catalog, it is found that at this energy concentration the maximum permissible furnace temperature is approximately 2300°F. Element life, of course, becomes relatively longer as the operating temperature is lowered.

Wiring. The heating elements are assembled in the grooves of the muffle and door, and the ends are brought out through small holes drilled through the back of the furnace and through the side of the door. The end of each wire is attached to a brass connector shown in Details *I* and *J*. These connectors, in turn, are assembled to a back panel made from a piece of asbestos sheathing, $15\frac{1}{2} \times 15\frac{1}{2}$ inches, and a door panel made from a piece 2×8 inches. These panels are attached to the furnace with No. 10 screws, 2 inches long, which pass through 1-inch spacers. The spacers may be made from $\frac{1}{2}$ -inch pipe or any other suitable material.

As shown in the wiring diagram, Detail *H*, connections are made to two single-pole, single-throw switches, and one double-pole, single-throw switch. The main line and connection wires are No. 10-gage copper, with the exception of the line running from the door coil to the panel: this is a length of No. 14-gage BX cable, which must be about 4 feet long to permit the door to be readily raised and lowered.

Door Supports. As shown in Figure 1, the door is raised vertically by a 4-foot length of woven-wire rope which passes over two pulleys to a counterweight. The rope is attached to an eye-bolt which is carried by a strap fastened to the exact center of the top of the door. When the door is lowered into position, it is tightly closed by means of two common window fasteners attached to the sides.

The dimensions of the pulleys and the supporting frame-work (Detail *K* of Figure 3) are of little importance although, obviously, it is necessary to line up the pulley so that the door will be lifted straight up and the weight will clear the side of the furnace. However, it is desirable to make an angle- or channel-iron guide for the counterweights, as shown in Figure 1. Although large sash weights were used as counterweights in the furnace illustrated, any other available weights may be used.

Precautions. Initial Firing. The first firing of the furnace should be carried out very slowly with the door partially open to drive the

moisture from the insulation and the firebricks. It has been found to be good practice to use only one set of coils for the initial firing, and then to turn off the power when the temperature reaches about 1000°F. and permit the furnace to stand overnight before further heating.

Care of the Heating Elements. Care should be exercised at all times to see that the heating elements are not bent or struck sharply, for heating makes them very brittle.

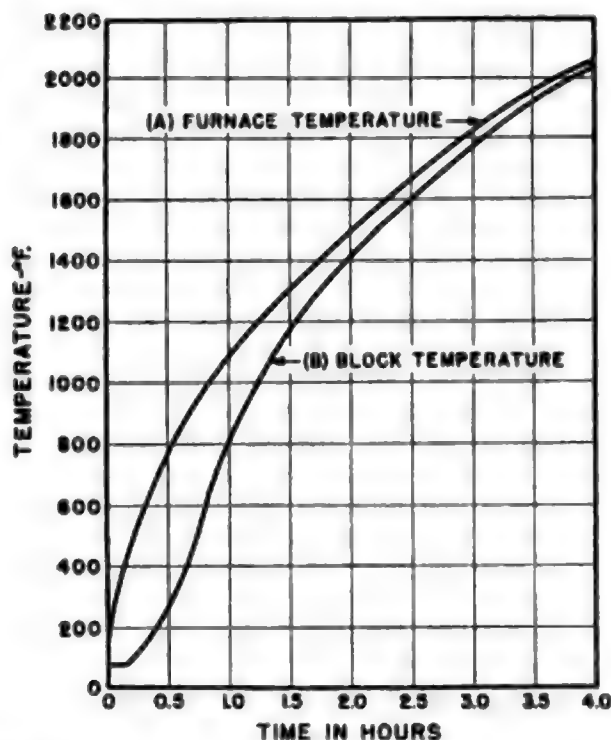


Figure 4. Rate of Temperature Rise in Muffle Furnace

The best protection for the bottom of the muffle is a $\frac{1}{2}$ - or $\frac{3}{8}$ -inch carborundum slab; this not only protects the wire but also prevents mechanical damage to the brick.

Performance. The furnace heats up fairly rapidly, as is shown by the time-temperatures curves of Figure 4. Curve A shows the temperature rise of the furnace itself; Curve B was obtained simultaneously with a thermocouple imbedded in the center of a 15.1-pound block of steel.

Pyrometer. An inexpensive and simple pyrometer for use up to temperatures of 2200°F. can be made with No. 8 gage chromel and alumel wires connected through a small resistance to a D.C. milliammeter. A satisfactory combination was found to be a milliammeter

with a range of 0 to 1 milliamperes and an external resistance of 12 ohms.*

Comments on Design. Furnaces of the size described here have proved to be very satisfactory for general laboratory work and for the heat treatment of metallurgical specimens. However, both smaller and larger furnaces have been built following the same principles of construction. Some of these have internal dimensions as small as 9 inches in width x 9 inches in depth x 5 inches in height; others are as large as 18 inches wide x 18 inches deep x 22½ inches high. The small furnaces require about 2400 watts while the larger ones, which are used to fire ceramic ware, require about 6600 watts.

SUGGESTED EXPERIMENT

Although this furnace was not designed for the study of heat transfer, furnaces of this type lend themselves readily to the study of heat transfer by conduction through resistances in series. A typical experiment involves the measurement of the power input required to maintain a definite furnace temperature, and a comparison of the actual power input with the value calculated from a knowledge of the thermal conductivities of the various insulating layers. An experiment of this type is made more valuable if thermocouples are embedded in the walls of the furnace at various points so that the actual temperature distribution can be compared with the values calculated from thermal conductivities.

The calculations involved in such an experiment are illustrated by the following problem:

Problem. To calculate: (1) the power required to maintain the furnace at 1800°F. with the room at 75°F., and (2) the temperature distribution through the walls.

Data. From the dimensions of the furnace (Figure 2), the mean area and the thickness of each layer were calculated; these results are given in Table 2 together with values of the thermal conductivities which were obtained from manufacturer's catalogs.

Power and Current Requirements. From the data in Table 2, the heat loss through the top and bottom, the sides, and the front and

* The milliammeter used was Model 321, Triplett Instrument Co., Bluffton, Ohio.

back of the furnace was calculated by means of the general equation for heat transfer by conduction:

$$\frac{Q}{\theta} = \frac{\Delta t}{\frac{L_1}{k_1 A_1} + \frac{L_2}{k_2 A_2} + \frac{L_3}{k_3 A_3} + \frac{1}{h_4 A_4}} \quad (1)$$

$$= \frac{\Delta t}{R_1 + R_2 + R_3 + R_4} = \frac{\Delta t}{R} \quad (2)$$

TABLE 2

Measurements and Thermal Conductivities of Insulating Layers of Muffle Furnace

Item	Insulating firebrick	Insulation block	Asbestos sheathing	Outside surface
Thermal conductivity, $k = \text{B.t.u.-ft.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.})$	0.16	0.067	0.43	—
Approximate air-film coefficient, $\text{B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.})$	—	—	—	2
Mean area, ft.^2 :				
Top and bottom.....	1.27	2.78	3.44	3.52
Sides.....	1.17	2.64	3.59	3.67
Front and back.....	1.36	2.92	3.59	3.67
Thickness, ft. :				
Top and bottom.....	0.208	0.333	0.0208	—
Sides.....	0.375	0.167	0.0208	—
Front and back.....	0.375	0.167	0.0208	—

where, in this particular case

$Q/\theta =$ heat lost, B.t.u./hr.

$\Delta t =$ temperature difference, furnace to air, $^\circ\text{F.}$

$L =$ thickness, ft.

$A =$ mean area, ft.^2

$k =$ thermal conductivity, $\text{B.t.u.-ft.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.})$.

$h =$ air film coefficient, $\text{B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.})$.

$R =$ thermal resistance, $(^\circ\text{F.} \times \text{hr.})/\text{B.t.u.}$

and the subscripts 1, 2, 3, and 4 refer to the insulation firebrick, the insulation block, the asbestos sheathing, and the outside air film, respectively.

1. *For the top and bottom*

$$\frac{Q}{\theta} = 2 \times \frac{1800 - 75}{\frac{0.208}{0.16 \times 1.27} + \frac{0.333}{0.067 \times 2.78} + \frac{0.0208}{0.43 \times 3.44} + \frac{1}{2 \times 3.52}}$$

$$\begin{aligned}
 &= 2 \times \frac{1725}{1.025 + 1.785 + 0.0141 + 0.1422} \\
 &= 2 \times \frac{1725}{2.9663} \\
 &= 1,164 \text{ B.t.u./hr.}
 \end{aligned}$$

2. *For the sides*

$$\begin{aligned}
 \frac{Q}{\theta} &= 2 \times \frac{1800 - 75}{\frac{0.375}{0.16 \times 1.17} + \frac{0.167}{0.067 \times 2.64} + \frac{0.0208}{0.43 \times 3.59} + \frac{1}{2 \times 3.67}} \\
 &= 2 \times \frac{1725}{2.005 + 0.943 + 0.0135 + 0.1364} \\
 &= 2 \times \frac{1725}{3.0979} \\
 &= 1,115 \text{ B.t.u./hr.}
 \end{aligned}$$

3. *For the front and back*

$$\begin{aligned}
 \frac{Q}{\theta} &= 2 \times \frac{1800 - 75}{\frac{0.375}{0.16 \times 1.36} + \frac{0.167}{0.067 \times 2.92} + \frac{0.0208}{0.43 \times 3.59} + \frac{1}{2 \times 3.67}} \\
 &= 2 \times \frac{1725}{1.725 + 0.853 + 0.0135 + 0.1364} \\
 &= 2 \times \frac{1725}{2.7279} \\
 &= 1,266 \text{ B.t.u./hr.}
 \end{aligned}$$

SUMMARY OF HEAT LOSSES FROM FURNACE:

1. From top and bottom = 1,164 B.t.u./hr.
 2. From sides = 1,115 B.t.u./hr.
 3. From front and back = 1,266 B.t.u./hr.
- Total 3,545 B.t.u./hr.

Therefore, the electrical energy required is

$$3,545 \times 0.000293 = 1.04 \text{ kw.-hr.}$$

where

$$0.000293 = \text{B.t.u./kw.-hr.}$$

and the current required to maintain the furnace at 1800°F. is approximately 10 amperes at 110 volts.

Temperature Distribution Through Walls. The temperature drop across each layer, Δt_i , is calculated from the overall temperature drop

TABLE 3

Temperature Drops through the Various Insulating Layers of the Muffle Furnace
[Temp. in °F.; resistance in (°F. × hr.)/B.t.u.]

Overall temp. drop and resistance	Insulating firebrick		Insulation block		Asbestos sheathing		Outside air film	
	R_1	Δt_1	R_2	Δt_2	R_3	Δt_3	R_4	Δt_4
Top and bottom: $\Delta t = 1725$ $R = 2.9663$	1.025	596	1.785	1,038	0.0141	8	0.1422	83
Sides: $\Delta t = 1725$ $R = 3.0979$	2.005	1,116	0.943	525	0.0135	8	0.1364	76
Front and back: $\Delta t = 1725$ $R = 2.7279$	1.725	1,090	0.853	540	0.0135	9	0.1364	86

TABLE 4

Temperature Distribution through Furnace Walls

Location	Temperature °F		
	Top and bottom	Sides	Front and back
Inside muffle wall.....	1800	1800	1800
Insulating firebrick—insulation block boundary....	1204	684	710
Insulation block—asbestos sheathing boundary....	166	159	170
Outside sheathing surface.....	158	151	161
Room.....	75	75	75

Δt , the overall thermal resistance R , and the resistance of the individual layer R_i , by means of the equation

$$\Delta t_i = \Delta t \times \frac{R_i}{R} \quad (3)$$

where $\Delta t_i = \Delta t_1, \Delta t_2, \Delta t_3$, etc.
and $R_i = R_1, R_2, R_3$, etc.

The various resistances and the calculated temperature drops are given in Table 3, and the temperature distribution through the furnace, calculated from the temperature drops, is given in Table 4.

NOMENCLATURE

A = area, ft.²

h = film coefficient, B.t.u./ (hr. \times ft.² \times °F.).

k = thermal conductivity, B.t.u.-ft./ (hr. \times ft.² \times °F.).

L = thickness, ft.

$\frac{Q}{\theta}$ = heat lost, B.t.u./hr.

R = thermal resistance, (°F. \times hr.)/B.t.u.

Δt = temperature difference, °F.

Subscripts

1 = insulation firebrick.

2 = insulation block.

3 = asbestos sheathing.

4 = outside air film.

l = individual layer.

SECTION 3

EVAPORATION

INTRODUCTION

Evaporation is a process for concentrating a solution by vaporizing part or all of the solvent—usually water. Essentially, evaporation may be considered to be a special case of heat transfer in which heat is transferred from condensing vapors, from hot gases, or directly by radiation to a liquid at a constant temperature, usually its boiling point at the operating pressure. By far the most common evaporators are those in which heat is transferred through a metal wall from condensing steam to a boiling liquid. In this case, the rate of heat transfer is expressed by the general equation:

$$Q/\theta = \frac{\Delta t}{\frac{1}{h_1 A_1} + \frac{L}{k A_m} + \frac{1}{h_2 A_2}} \quad (3.1)$$

$$= U_1 A_1 \Delta t = U_m A_m \Delta t = U_2 A_2 \Delta t \quad (3.2)$$

where

Q/θ = rate of heat transfer, B.t.u./hr.

Δt = temperature drop across the heating surface, °F.

A_1 , A_m , and A_2 = area of the inner, the mean, and the outer heating surface, respectively, ft.²

k = thermal conductivity of the metal wall, B.t.u.-ft./
(hr. \times ft.² \times °F.).

h_1 and h_2 = inner- and outer-film coefficient, respectively,
B.t.u./(hr. \times ft.² \times °F.).

U_1 , U_m , and U_2 = overall heat-transfer coefficient based on the inner, the mean, and the outer heating area, respectively,
B.t.u./(hr. \times ft.² \times °F.).

Overall Coefficients. In evaporation, as in other cases of heat transfer, the preferable practice is to base calculations on the individual film coefficients, but the use of overall coefficients is much more common. The situation is even further complicated by the fact that the overall coefficient may be either an apparent or a corrected coefficient.

Apparent Overall Heat-Transfer Coefficient. In evaporator practice it is customary to calculate the temperature of the boiling liquid from

the pressure in the vapor space, neglecting such factors as superheat, elevation of boiling point, and hydrostatic head. The temperature drop determined in this manner is the difference in temperature between the boiling points of water at the pressure on the steam side of the heating surface and in the vapor space above the boiling liquid. The overall coefficient calculated on the basis of this temperature drop is known as the apparent coefficient because it is based on an apparent, but not necessarily a real, temperature difference.

Corrected Overall Coefficients. Apparent coefficients are frequently very misleading because the temperature calculated from the pressure in the vapor space is often much lower than the true temperature of the liquid. There are two reasons for this: (1) a solution boils at a higher temperature than the pure solvent, and (2) the boiling point of a pure liquid calculated from the pressure of the vapor above it represents only the temperature at the surface of the liquid, for every particle of liquid below the surface is at a pressure equal to the sum of the pressure in the vapor space plus the pressure due to the hydrostatic head of liquid above the particle; therefore, it boils at a temperature higher than the temperature at the surface. If the apparent temperature drop is corrected for the elevation of boiling point, the overall coefficient calculated on this basis is known as the *overall heat transfer coefficient corrected for elevation of boiling point*; if the apparent temperature drop is also corrected for the effect of hydrostatic head, the coefficient is the *overall coefficient corrected for elevation of boiling point and hydrostatic head*.

Elevation of Boiling Point. The magnitude of the elevation of boiling point of a solution depends upon the molecular or ionic concentration of the dissolved particles. For colloidal systems, the effect is usually negligible because even though the total concentration of solids may be high the individual particles are large and their number is relatively small. With salts, acids, and alkalies the elevation of boiling point may be very high because of the small size—and, therefore, the large number—of molecules and ions.

A very useful method for correlating and estimating vapor pressure and boiling point elevation data is that expressed by *Dühring's rule*, which states that the temperature at which one liquid exerts a given vapor pressure is a linear function of the temperature at which a second liquid exerts the same pressure.¹⁰ According to this rule, if the boiling points of a liquid are plotted against the boiling points at corresponding pressures of a second liquid, a straight line results. In evaporation work, it is most convenient to plot the boiling points of solutions against the boiling point of water, as in Figures 1 and 2, which show the Dühr-

ing lines for sodium chloride and sodium hydroxide solutions. As only two points are needed to plot a Dühring line, this method is par-

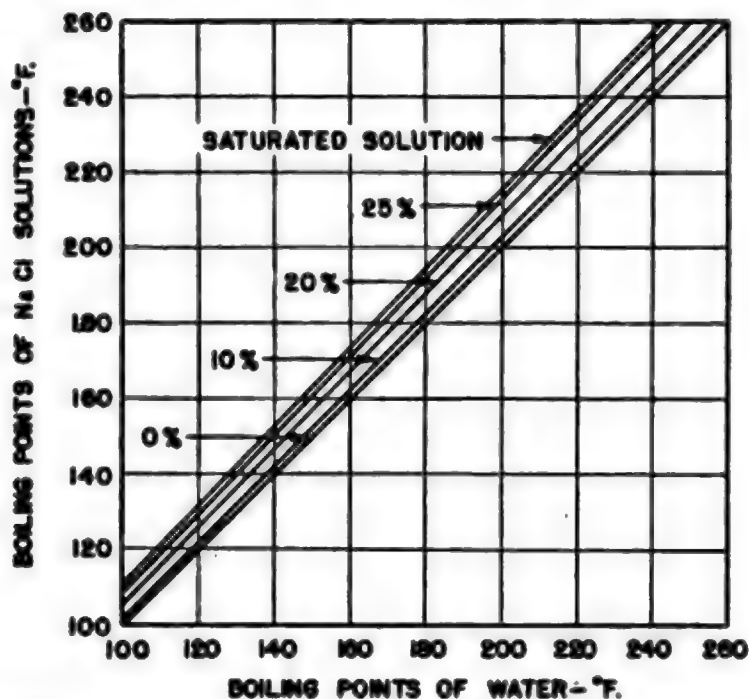


Figure 1. Dühring Lines for Sodium Chloride Solutions

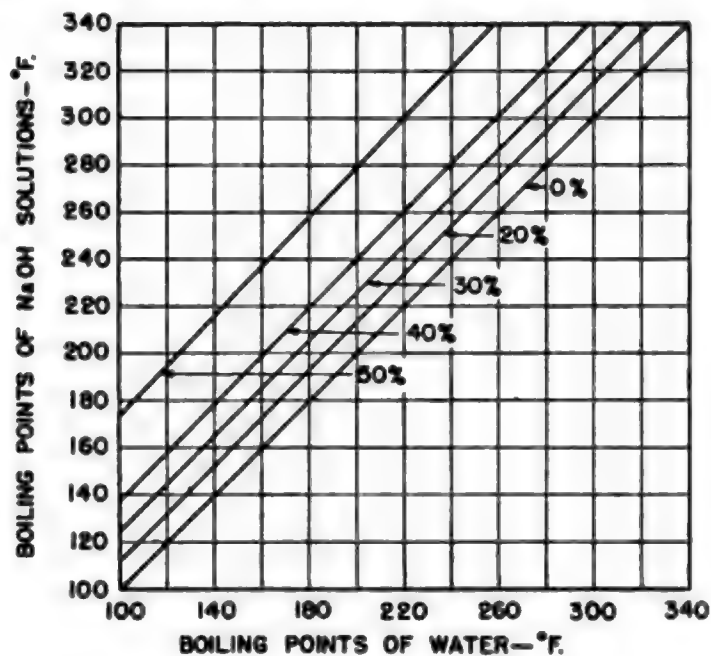


Figure 2. Dühring Lines for Sodium Hydroxide Solutions

ticularly valuable for interpolating and extrapolating vapor pressure data when the values at only two temperatures are known.

Hydrostatic Head. Overall coefficients are seldom corrected for the effect of hydrostatic head because such corrections cannot be made accurately. Due to the violent agitation and circulation of the liquid, the temperature at any point cannot be predicted; cooler liquid from the surface is forced to the bottom where the pressure is greater than its vapor pressure, and as it rises it first becomes heated until it boils and then becomes progressively cooler as it flashes with decreasing pressure. The simplest way to correct for hydrostatic head is to assume that the mean temperature is half way between the top and bottom of the liquid; this may not be accurate, but it is better than entirely neglecting the effect of hydrostatic head.⁴

Types of Evaporators. Many types of evaporators have been built, but most of these have had rather limited application. At present, the most important are the horizontal-tube type with steam inside the tubes, and four vertical-tube types—the standard, the basket-type, the long tube, and the forced circulation—in all of which the steam is outside the tubes.

In general, horizontal-tube evaporators are satisfactory for non-viscous solutions that do not deposit scale or salt; standard or basket-type vertical tube evaporators are preferred for liquids of somewhat higher viscosities and where scale or salt forms during evaporation; long-tube evaporators are desirable where the viscosity is reasonably high and scaling or salting does not occur;^{1, 6} forced-circulation evaporators are particularly suited for liquids of high viscosity or those which tend to foam or to deposit scale or salt on the heating surfaces.^{2, 4, 7, 8, 9}

Multiple-Effect Operation. In practice, from two to five individual evaporators are usually operated in series to form a multiple-effect evaporator; in such installations the vapors from one effect are the heating medium for the next succeeding effect, and only the vapors from the last effect go to the condenser. This results in considerable saving of steam for, theoretically, each pound of steam admitted to the system will bring about the evaporation of approximately one pound of water from each effect; however, radiation losses and sensible heat effects may appreciably lower the economy.

The disadvantage of multiple-effect evaporation is that it requires more equipment than single-effect operation, although this is usually offset by the greater economy.

In an idealized system in which there were no heat losses or sensible heat effects and in which all effects had the same areas and coefficients,

the capacity of a multiple-effect evaporator would be the same as the capacity of each of the individual evaporators operated as single-effects under the same conditions of steam pressure and pressure in the condenser. For, if

Q/θ = heat transferred per effect.

A = area of the heating surface in each effect.

U = overall heat transfer coefficient of each effect.

N = number of effects.

Δt = overall temperature drop, steam to condenser.

the temperature drop across each effect is $\frac{\Delta t}{N}$, and the heat transferred in each effect is

$$Q/\theta = \frac{UA\Delta t}{N} \quad (3.3)$$

and the total heat transferred is

$$N \times Q/\theta = N \times \frac{UA\Delta t}{N} \quad (3.4)$$

$$= UA\Delta t \quad (3.5)$$

which is the same regardless of the value of N .

Actually, the capacity of a multiple-effect evaporator is less than the capacity of each individual evaporator operated as a single. This is due to four factors: (1) increased radiation losses, (2) boiling point elevation, (3) hydrostatic head, and (4) variation in the overall coefficient.

Radiation losses, obviously, increase as the exposed surface increases, but this is not particularly serious for radiation losses can be kept to a minimum with proper insulation; the other factors, however, are very important. Wherever there is an elevation of boiling point or a hydrostatic head, the difference between the temperature of the liquid and the temperature corresponding to the pressure above the liquid represents a loss in available temperature drop. When there is an elevation of boiling point, the vapors leave the liquid surface in a superheated condition, but they give up their heat in the next effect at their condensing temperature; the loss due to hydrostatic head occurs through flashing beneath the liquid surface. In a single-effect, such loss in available temperature drop occurs only once; in a multiple-effect it occurs in each effect, and the sum of such losses in multiple-effect operation represents an appreciably greater percentage of the available temperature drop than it does in single-effect operation.

In multiple-effect operation, the overall coefficients are not the same in each effect, but decrease as both the boiling point and the temperature drop decrease³; this is due to the increase in the viscosity of the films. As the viscosity of the films is also influenced by the concentration of the dissolved solids, the coefficient will also be different in each effect because of concentration differences. Therefore, even if there were no elevation of boiling point or similar effects, a multiple-effect evaporator could hardly be expected to have the same capacity as a single-effect.

Operation Under Vacuum. It is obvious that no heat can be transferred unless there is a drop in temperature, and in evaporation, where heat is transferred from condensing steam to a boiling solution, this means that there must be a corresponding drop in pressure. From a practical point of view, this necessitates operation under vacuum, for in many cases multiple-effect operation would be impossible at the steam pressures available if vacuum were not used, while, in any case, the use of vacuum means an increased temperature drop and a corresponding decrease in the required heating surface. For certain organic materials, vacuum operation is also essential to minimize decomposition by reducing the temperature.

Operation under vacuum requires the use of a pump to eliminate non-condensable gases from the system. When this is done, the pressure at each point in the system will be the vapor pressure of water at its corresponding temperature. Therefore, the lowest pressure attainable is determined by the temperature of the water in the condenser. Most evaporators are provided with some type of spray condenser in which the vapors from the last effect come in intimate contact with the condenser water. Such condensers are not only the lowest in cost, but the intimate contact of the vapor and liquid results in lower water consumption for a given condenser temperature than is possible in the more expensive surface condenser. However, where it is desired to collect the condensate without contamination with condenser water, surface condensers must be used.

Methods of Feeding. A multiple-effect evaporator may be operated with forward, backward, or parallel feed, or with a combination of feeds. In forward feed, the feed is introduced into the first, or highest-temperature effect, and the liquor flows through the remaining effects, becoming more concentrated in each. In backward feed, flow is in the opposite direction: the feed is introduced into the last effect, and is then pumped through the others in series. In parallel feed, all effects

are fed with the raw feed and no liquor is transferred from one effect to the other.

Forward feed is most common for it is easy to control, and the pumping arrangement is very simple: only a pump to feed the first effect and another to remove thick liquor from the last effect are required. If the feed is hot, this method is economical, but if the feed is cold, it is heated to boiling temperature by live steam in the first effect and this causes an increase in the overall steam consumption per pound of water evaporated, for steam condensed in heating the feed forms no vapor for use in the next effect.

Backward feed has certain advantages and disadvantages over forward feed. If the feed is hot, the steam consumption is lower with forward feed than with backward feed, but if the feed is cold, backward feed is more economical, for the feed is heated to boiling temperature by vapors which have resulted from previous evaporation instead of by live steam, which would be the case if the cold feed were introduced into the first effect. However, the particular value of backward feed is in its influence on the overall heat-transfer coefficients. In forward feed, the most concentrated liquor is in the effect where the temperature is the lowest, and the combination of high concentration and low temperature results in high viscosity, which may be reflected in excessively low heat-transfer coefficients in the last effect, and low overall evaporator capacity. In backward feed, the highest concentration is in the effect where the temperature is highest; this decreases the viscosity of the liquor in the last effect and results in a higher coefficient, which is reflected in a higher overall evaporator capacity. The principal disadvantage of backward feed is that a pump is required between each effect, and control of the liquor levels in the effects is somewhat more difficult than with forward feed.

Parallel feed is not as common as the other methods of feeding, but it is the most convenient method where a saturated feed is being evaporated and crystallized and no thick liquor is withdrawn from the evaporator.

Laboratory Evaporators. Laboratory evaporators should be small-scale models of the larger commercial units; they should not be so large that they require excessive amounts of steam and materials, and they should not be so small that the results are not indicative of large-scale operation. Suitable evaporators may be purchased from manufacturers, they may be built to specifications by fabricating companies, or they may be built in the laboratories themselves. In any case, they should be constructed with proper devices and facilities so that such

data as temperatures, pressures, concentrations, and quantities of steam, cooling water, and condensates can be readily obtained. For laboratory purposes, a multiple-effect evaporator is much better than a single-effect, for it is representative of the most common commercial practice. Furthermore, a multiple-effect evaporator permits studies with forward, backward, and parallel feed and thus enables investigations to be made under a variety of conditions.

A laboratory evaporator may be built of identical units, or it may be a combination of different types; for example, a basket-type, a long-tube, and a forced-circulation evaporator arranged to be operated with all units as singles, any two as a double, or all three as a triple is a very good setup.

Evaporator Calculations. Most problems in evaporation deal with the calculation of heating areas when the coefficients are known. Most laboratory work in evaporation, however, is concerned with determining the heat-transfer coefficients when the areas are known. In either case, the first step in the solution of the problem is to write a heat balance around each effect. In writing these heat balances such quantities as superheat, heats of solution and dilution, and radiation losses are frequently disregarded. In some cases this is done because the data are not available; in other cases it is felt that the magnitude of these factors is small compared to the latent heat of the vapor and the sensible heat of the liquid. Nevertheless, it is advisable to make certain that these effects are really negligible, for sometimes they are quite important. Typical calculations are illustrated on pages 232-238.

NOMENCLATURE

- A = area of heating surface, ft.²
- h = film coefficient, B.t.u./(hr. x ft.² x °F.).
- k = thermal conductivity, B.t.u.-ft./(hr. x ft.² x °F.).
- L = wall thickness, ft.
- N = number of effects.
- Q = heat transferred, B.t.u.
- Δt = temperature drop across heating surface, ft.²
- U = overall heat-transfer coefficient, B.t.u./(hr. x ft.² x °F.).
- θ = time, hr.

Subscripts

- 1 = inner.
- 2 = outer.
- m = mean.

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A TRIPLE-EFFECT EVAPORATOR

Designed and Constructed by

The Departments of Chemical Engineering
of The University of North Dakota
and The University of New Hampshire*

Description. This evaporator has a number of features which make it particularly suitable as an experimental unit: (1) it resembles a large-scale evaporator in all essential details; (2) it has high capacity; (3) it is flexible in operation and can be used for a variety of different evaporating conditions; and (4) it can be easily disassembled for cleaning or repairing.

The three bodies of the evaporator are identical: they are made from flanged 5-foot sections of standard 10-inch steel pipe, and each is provided with a conical bottom and a salt catcher so that the system can be used for evaporation where crystallization occurs. The heating units are baskets, each of which is made from a 2-foot length of 6-inch copper tubing and 18 lengths of $\frac{3}{4}$ -inch copper tubing, silver-soldered in place; the heating surface in each effect is approximately 11.3 square feet.

Instead of the customary spray or jet condenser, this evaporator has a surface condenser to permit the collection of the condensate in calibrated receivers; this condenser has 4 passes and a total condensing area of about 43.6 square feet. Provision is also made for cooling and collecting the condensate from each of the heating units. Duplicate receivers are provided so that one can be emptied while the other is in use.

As the evaporator was designed for experimental purposes, it was arranged in such a manner as to permit: (1) the first effect to be operated as a single; (2) the first and second effects to be operated together as a double with forward, backward, or parallel feed; and (3) all three effects to be operated as a triple with forward, backward, or parallel feed.

Figure 1 is a photograph of the evaporator at the *University of North Dakota*; Figures 2 to 6 show the construction of the *University of New Hampshire* evaporator.

Materials and Cost. The materials used in constructing the evaporator assembly are listed in Table 1. The cost of the complete assembly, exclusive of labor, was approximately \$610.

* Original design by the University of North Dakota; modified slightly by the University of New Hampshire.

and its volume V_p would be:

$$V_p = D^3 \quad (2)$$

Although the particles are not cubes, it can be assumed that on the average their areas and volumes are proportional to the areas and vol-

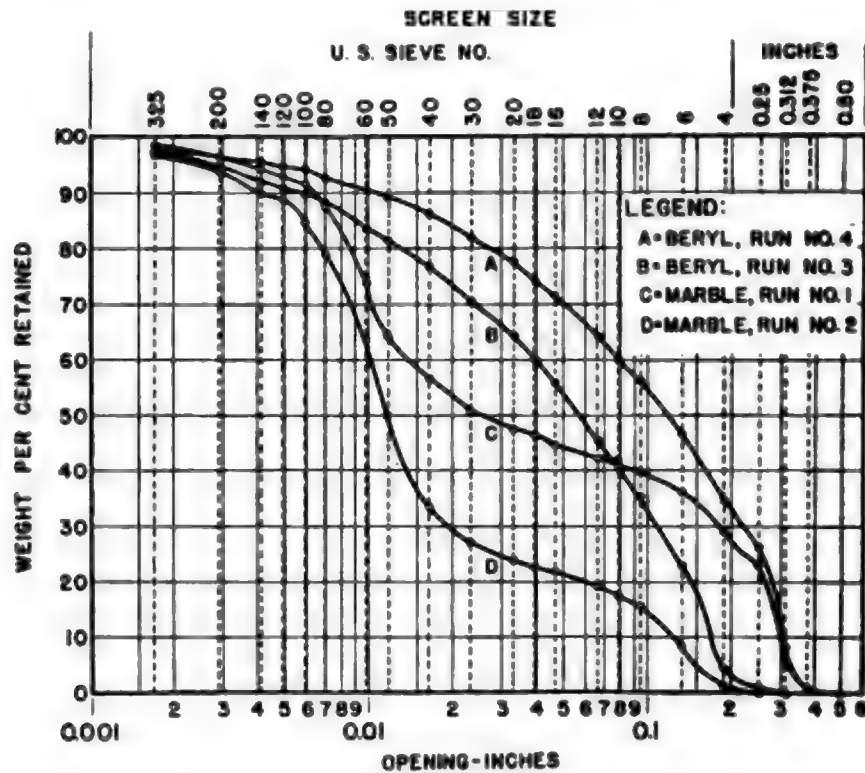


Figure 3. Screen Analyses of Crushed Marble and Beryl

umes of cubes of the same screen size; or

$$A_p = aD^2 \quad (3)$$

and

$$V_p = bD^3 \quad (4)$$

where a and b are constants for a given material.

The number of particles N in a uniform-sized fraction of weight w is

$$\begin{aligned} N &= \frac{w}{\rho} \times \frac{1}{V_p} \\ &= \frac{w}{\rho} \times \frac{1}{bD^3} \\ &= \frac{w}{cD^3} \end{aligned} \quad (5)$$

where ρ is the density, and c equals $b\rho$.

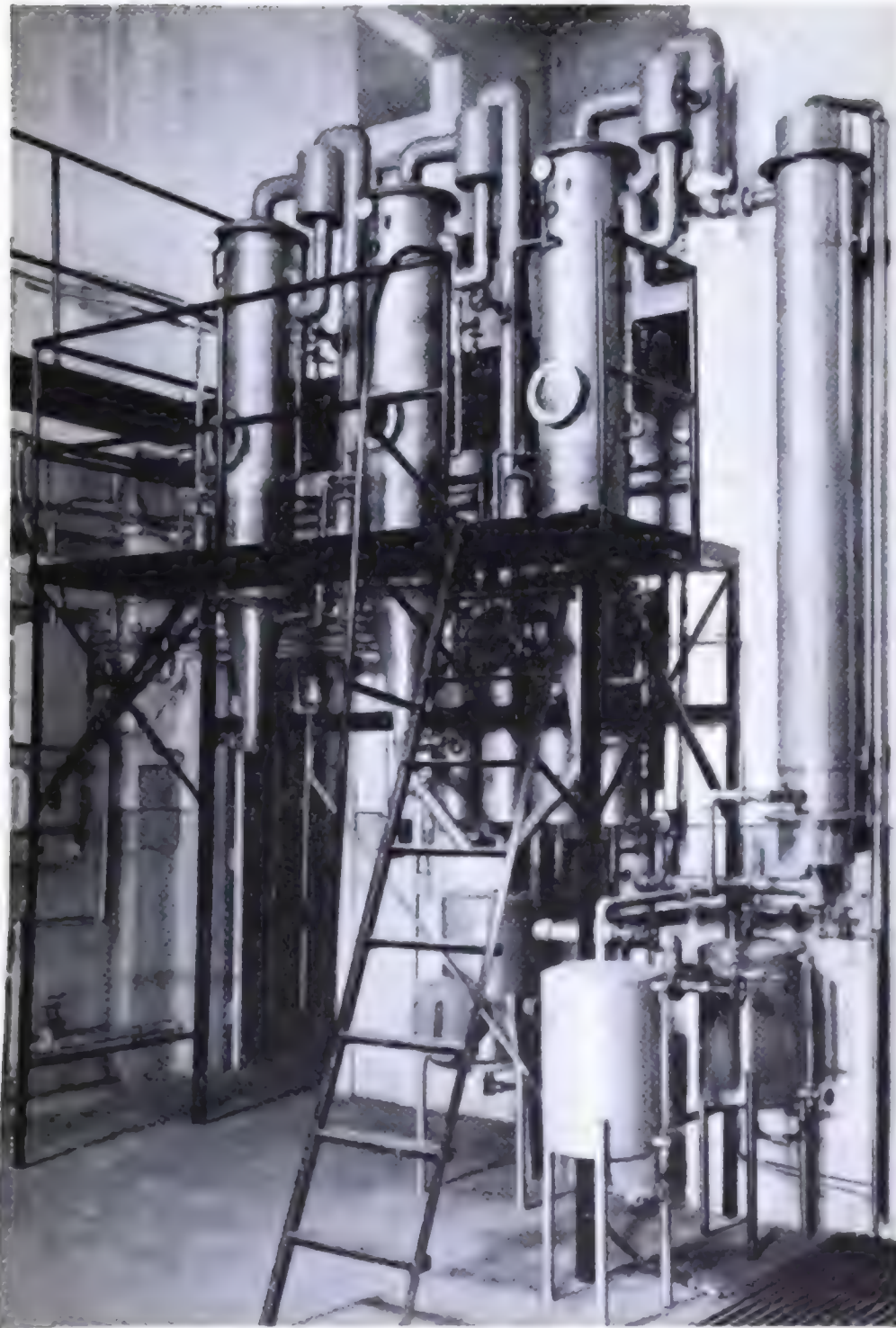


Figure 1. Triple-Effect Evaporator—University of North Dakota

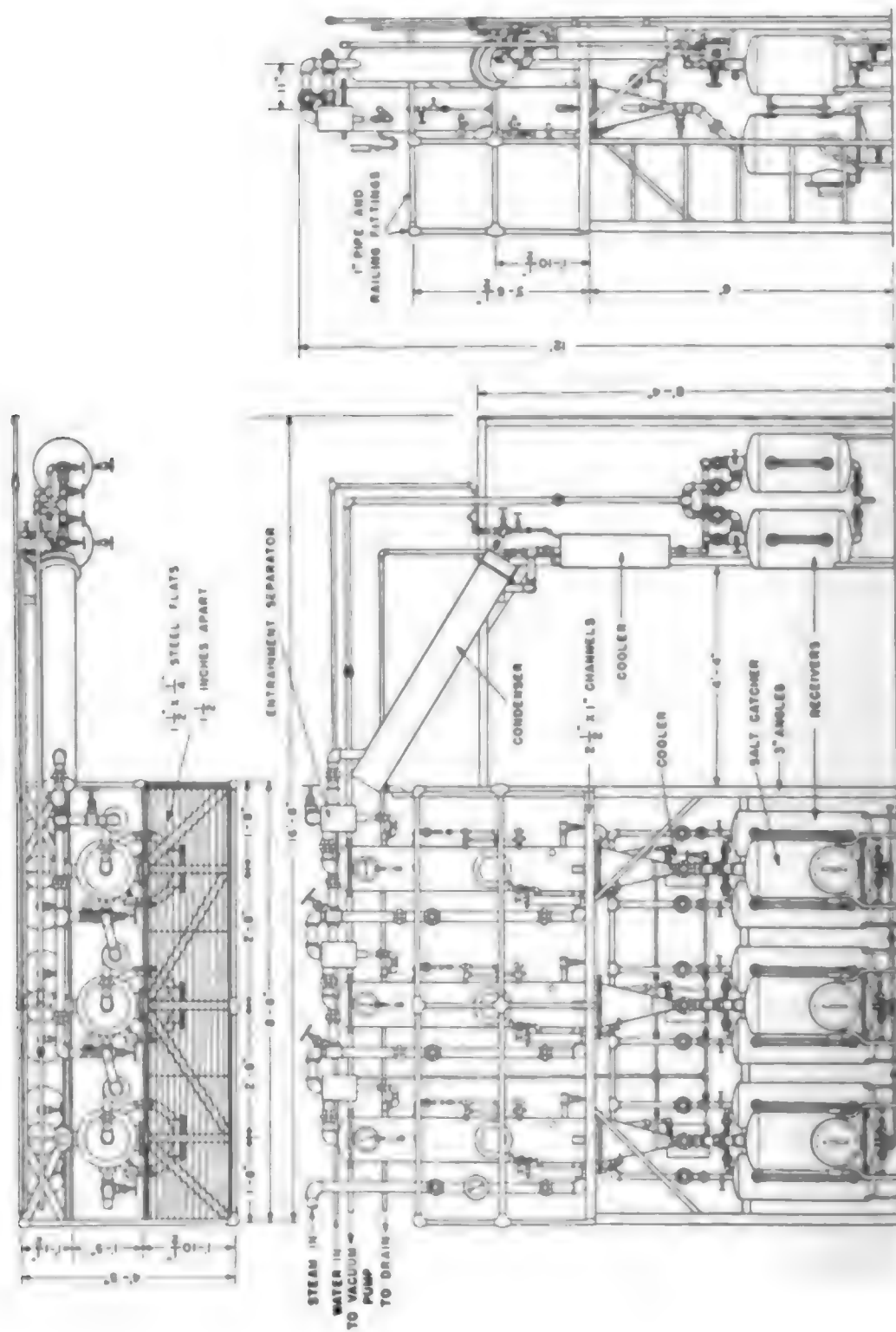


Figure 2 Assembly of Triple-Effect Evaporator—University of New Hampshire

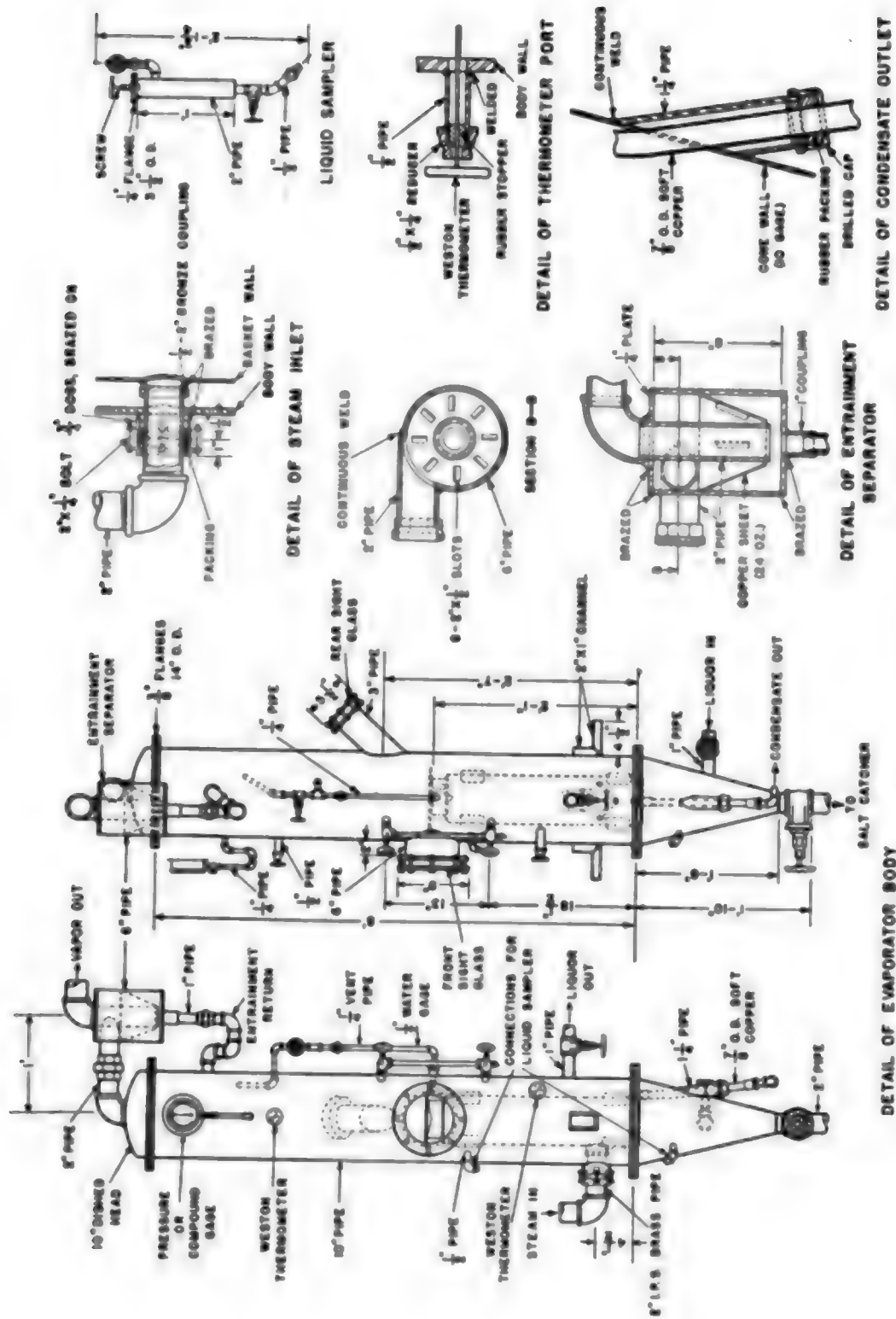


Figure 3. Details of Construction of Evaporator Body

Notes on Construction. Bodies. The 10-inch pipes for the evaporator bodies (Figure 3) were first fitted with flanges; these flanges, like all

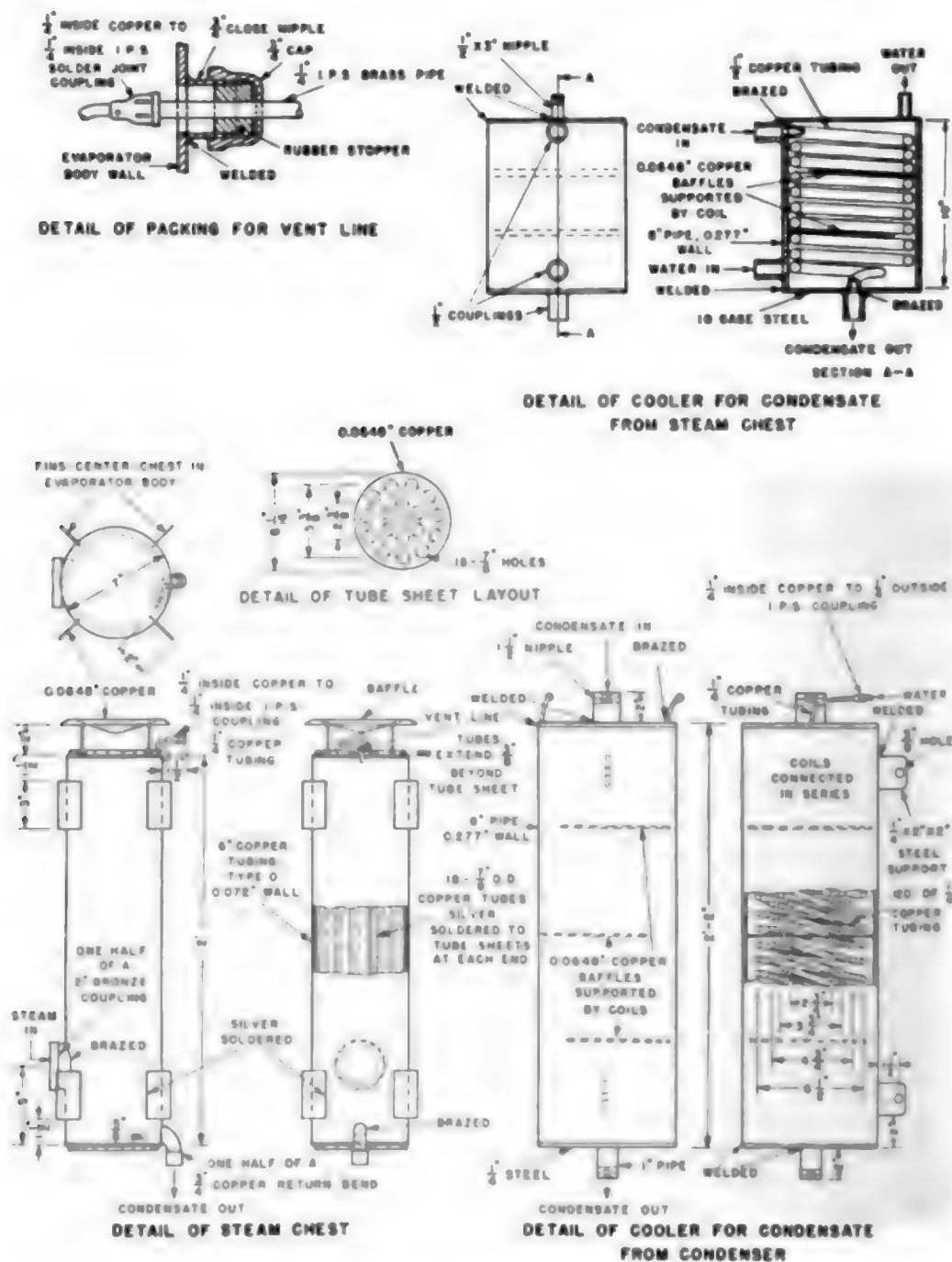


Figure 4. Details of Construction of Coolers and Steam Chests

other flanges for the evaporator, were cut with an acetylene torch from $\frac{3}{8}$ -inch steel plate, and finished to size on a lathe. The bolt holes were drilled, and the flanges were then welded in place. To minimize the

effect of warping during the welding of the flanges, they were assembled in pairs with a blank $\frac{3}{8}$ -inch steel plate between them.

After the flanges were fitted to the bodies, the various openings for the sight glasses, water gages, vents, steam and vapor inlets, ther-

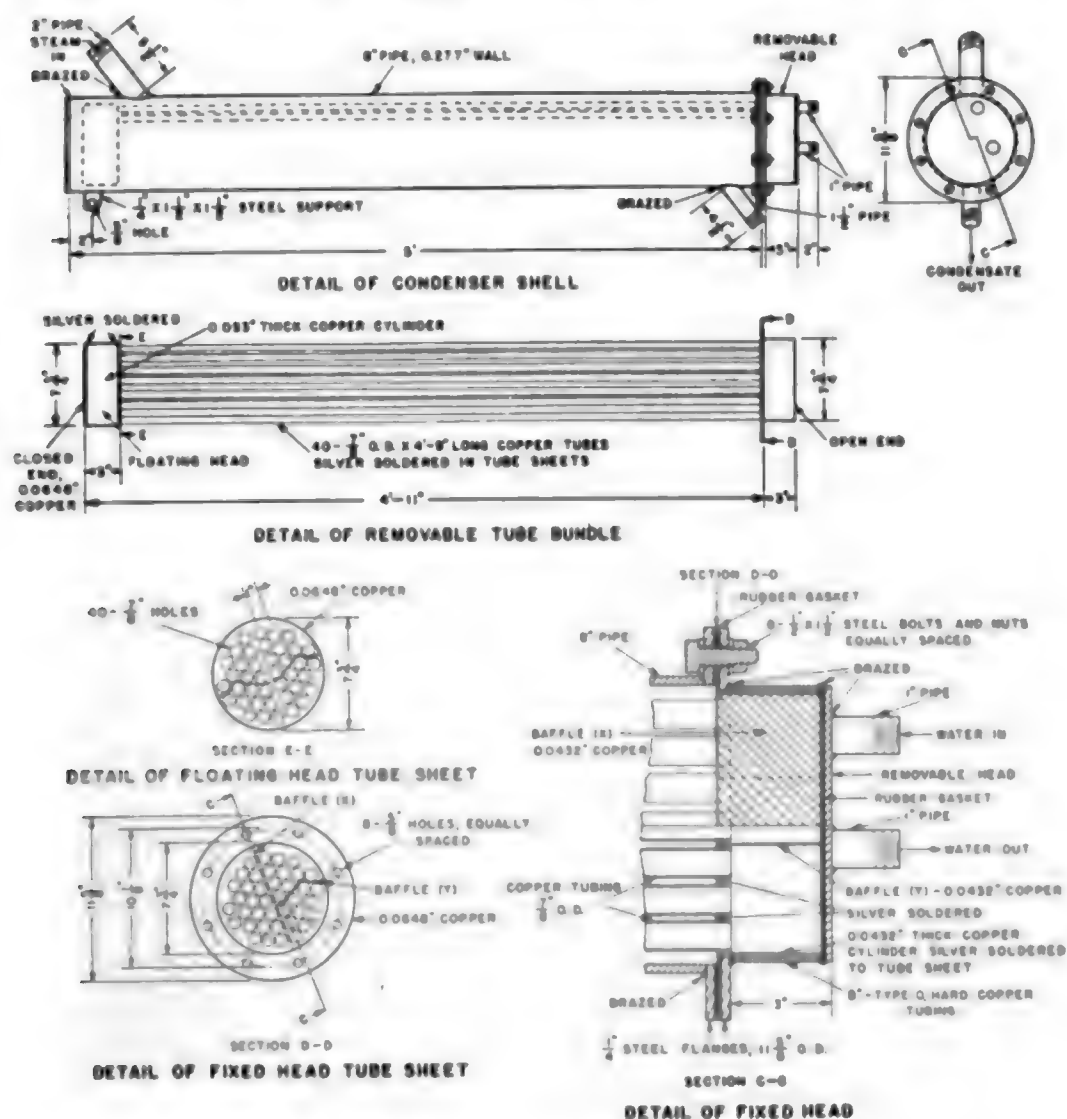


Figure 5. Details of Construction of Condenser

mometers, pressure gages, and liquid samplers were cut with a torch. The various nipples, supporting angles, and sight glass assemblies were then welded in place; the latter were fitted with flanges in the same manner as the evaporator bodies themselves.

Conical bottoms. The conical bottoms of the *University of North Dakota* evaporator were fabricated to order from 10-gage steel plate by a

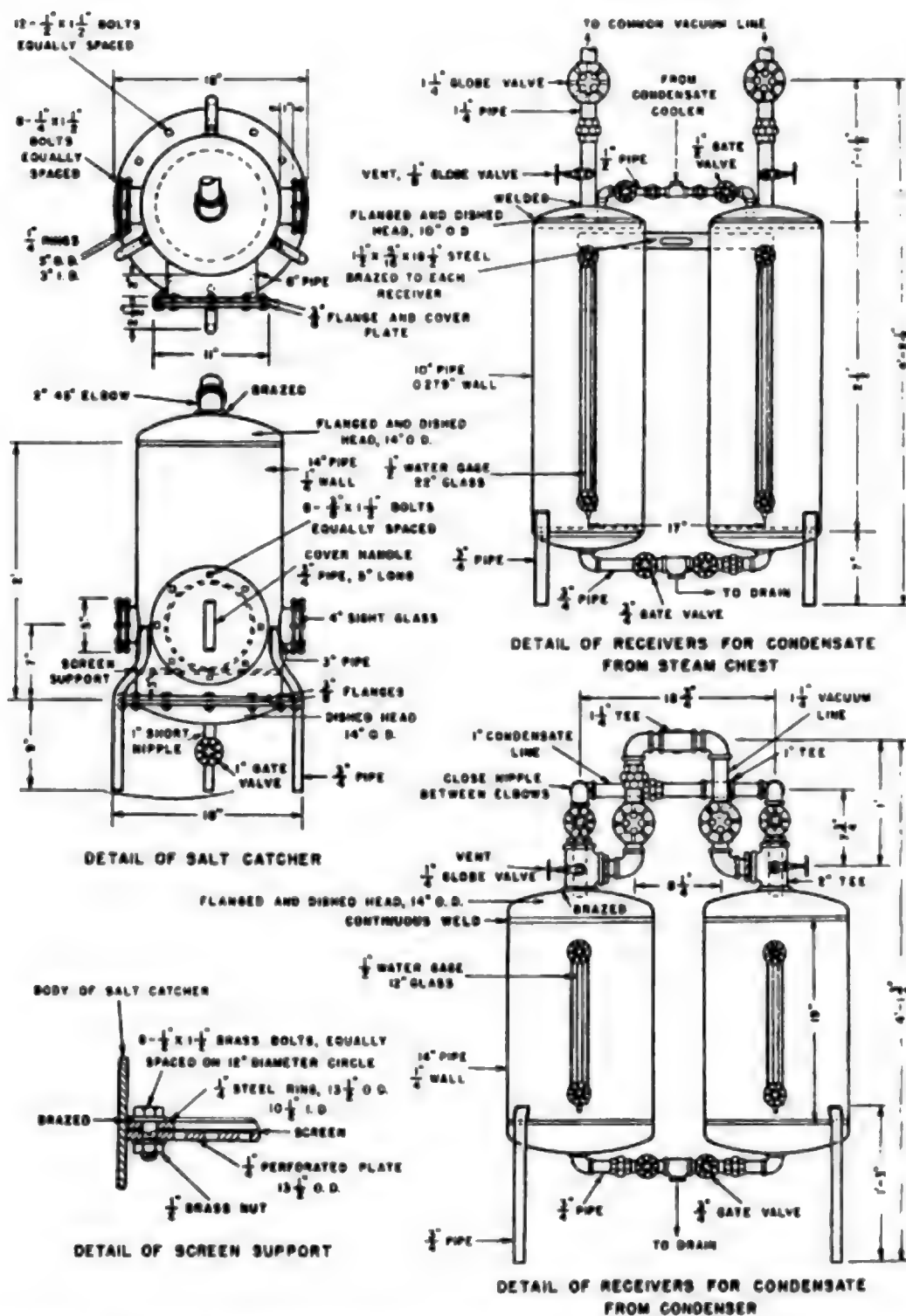


Figure 6. Details of Construction of Condensate Receivers and Salt Catchers

TABLE 1
Bill of Material for Triple-Effect Evaporator
 University of New Hampshire

Quantity	Description	Size	Material
6 pieces	Pipe	14 in. O.D. x 13.5 in. I.D. x 2 ft. long	Black wrought steel
2 pieces	"	14 in. O.D. x 13.5 in. I.D. x 18 in. long	" " "
3 pieces	Std. pipe	10 in. x 0.279 in. thick x 5 ft. long	" " "
6 pieces	" "	10 in. x 0.279 in. thick x 2½ ft. long	" " "
1 piece	" "	8 in. x 8½ in. long	" " "
1 piece	" "	6 in. x 26 in. "	" " "
1 piece	" "	6 in. x 2 ft. "	" " "
27 in.	" "	3 in.	" " "
16½ ft.	" "	2 in.	" " "
38 ft.	" "	1½ in.	" " "
175 ft.	" "	1 in.	" " "
44½ ft.	" "	¾ in.	" " "
9½ ft.	" "	½ in.	" " "
3 ft.	" "	¼ in.	" " "
12	Std. nipple	2 in. x 6 in. long	" " "
1	" "	2 in. x 5 in. "	" " "
6	" "	2 in. x 4½ in. "	" " "
4	" "	2 in. x 3 in. "	" " "
9	" "	2 in., short	" " "
5	" "	2 in., close	" " "
1	" "	1½ in. x 6 in. long	" " "
1	" "	1½ in. x 5 in. "	" " "
1	" "	1½ in., short	" " "
2	" "	1½ in., close	" " "
9	" "	1½ in. x 6 in. long	" " "
2	" "	1½ in. x 5½ in. "	" " "
1	" "	1½ in. x 5 in. "	" " "
6	" "	1½ in. x 2½ in. "	" " "
1	" "	1½ in. x 2 in. "	" " "
3	" "	1½ in., short	" " "
5	" "	1½ in., close	" " "
1	" "	1 in. x 5½ in. long	" " "
2	" "	1 in. x 4 in. "	" " "
4	" "	1 in. x 3½ in. "	" " "
3	" "	1 in. x 3 in. "	" " "
4	" "	1 in. x 2½ in. "	" " "
9	" "	1 in. x 2 in. "	" " "

TABLE 1—(Continued)

Quantity	Description	Size	Material
13	Std. nipple	1 in., short	Black wrought steel
10	" "	1 in., close	" " "
1	" "	$\frac{1}{2}$ in. x $4\frac{1}{2}$ in. long	" " "
4	" "	$\frac{1}{2}$ in. x $3\frac{1}{2}$ in. "	" " "
2	" "	$\frac{1}{2}$ in. x 3 in. "	" " "
3	" "	$\frac{1}{2}$ in. x 2 in. "	" " "
3	" "	$\frac{1}{2}$ in. x $1\frac{1}{2}$ in. "	" " "
22	" "	$\frac{1}{2}$ in., close	" " "
2	" "	$\frac{1}{2}$ in. x 6 in. long	" " "
4	" "	$\frac{1}{2}$ in. x 5 in. "	" " "
10	" "	$\frac{1}{2}$ in. x $2\frac{1}{2}$ in. "	" " "
20	" "	$\frac{1}{2}$ in. x 2 in. "	" " "
13	" "	$\frac{1}{2}$ in., short	" " "
45	" "	$\frac{1}{2}$ in., close	" " "
1	" "	$\frac{1}{2}$ in. x 5 in. long	" " "
3	" "	$\frac{1}{2}$ in. x 4 in. "	" " "
3	" "	$\frac{1}{2}$ in. x $3\frac{1}{2}$ in. "	" " "
4	" "	$\frac{1}{2}$ in. x 2 in. "	" " "
14	" "	$\frac{1}{2}$ in. x $1\frac{1}{2}$ in. "	" " "
6	" "	$\frac{1}{2}$ in., close	" " "
6	" "	$\frac{1}{2}$ in., close	" " "
8	Std. tee	2 in.	Black malleable iron
7	" "	$1\frac{1}{2}$ in.	" " "
15	" "	1 in.	" " "
11	" "	$\frac{3}{4}$ in.	" " "
9	" "	$\frac{1}{2}$ in.	" " "
4	" "	$\frac{1}{2}$ in.	" " "
20	Std. 90° elbow	2 in.	" " "
3	" " "	$1\frac{1}{2}$ in.	" " "
8	" " "	$1\frac{1}{2}$ in.	" " "
41	" " "	1 in.	" " "
19	" " "	$\frac{3}{4}$ in.	" " "
19	" " "	$\frac{1}{2}$ in.	" " "
22	" " "	$\frac{1}{2}$ in.	" " "
7	" 45° "	$\frac{1}{2}$ in.	" " "
13	Ground-joint union	2 in.	" " "
1	" " "	$1\frac{1}{2}$ in.	" " "
11	" " "	$1\frac{1}{2}$ in.	" " "
29	" " "	1 in.	" " "
11	" " "	$\frac{3}{4}$ in.	" " "
21	" " "	$\frac{1}{2}$ in.	" " "
5	" " "	$\frac{1}{2}$ in.	" " "
3	Std. coupling	1 in.	" " "
4	" "	$\frac{1}{2}$ in.	" " "

TABLE 1—(Continued)

Quantity	Description	Size	Material
3	Std. cap	1½ in.	Black malleable iron
3	" "	¾ in.	" " "
3	" "	½ in.	" " "
1	Std. reducer	1 x ¾ in.	" " "
2	" "	1 x ½ in.	" " "
6	" "	½ x ½ in.	" " "
2	Bushing	2 x 1½ in.	Cast iron
2	"	2 x 1 in.	" "
1	"	1 x ½ in.	" "
2	"	1 x ½ in.	" "
1	"	¾ x ½ in.	Black malleable iron
1	Plug, square head	2 in.	Cast iron
1	" " "	1 in.	" "
2	" " "	¾ in.	" "
1	" " "	½ in.	Steel
9	Std. gate valve	2 in.	Brass
19	" " "	1 in.	"
8	" " "	¾ in.	"
16	" " "	½ in.	"
1	" " "	½ in.	"
8	Std. globe valve	1½ in.	"
5	" " "	½ in.	"
6	" " "	½ in.	"
6	Railing tee, plain pattern	1 in.	Black malleable iron
5	Railing 90° elbow, plain pattern	1 in.	" " "
1	Railing cross, plain pattern	1 in.	" " "
6	Water gage	½ in.; 20 in. glass	Brass
5	" "	½ in.; 12 in. glass	"
1	Steam-gage siphon	½ in.	Black wrought steel
1	Pressure gage	0 to 100 lb., 3½ in. face	Brass
1	Compound gage	30 lb. press. and 30 in. vac.; 3½ in. face	"
1	" "	15 lb. press. and 30 in. vac.; 3½ in. face	"
6	Weston bi-metal thermometer	50 to 300°F., in 2°F.	Stainless steel
1	Water meter	½ in.; graduated in 0.01 cu. ft.	Bronze
57 ft.	Channels	2½ x 1½ x ½ in.	Steel

TABLE 1—(Concluded)

Quantity	Description	Size	Material
22 ft.	Channels	2 x $\frac{1}{2}$ x $\frac{1}{2}$ in.	Steel
34 ft.	"	2 x 1 x $\frac{1}{8}$ in.	"
26 ft.	Angles	3 x 3 x $\frac{1}{2}$ in.	"
8 ft.	"	1 $\frac{1}{2}$ x 1 $\frac{1}{2}$ x $\frac{1}{8}$ in.	"
44 ft.	"	1 $\frac{1}{2}$ x 1 $\frac{1}{2}$ x $\frac{1}{8}$ in.	"
120 ft.	Flats	$\frac{1}{8}$ x 1 $\frac{1}{2}$ in.	"
10	Flanged and dished head	14 in. O.D. x No. 9 gage	"
15	Flanged and dished head	10 in. O.D. x No. 9 gage	"
1	Sheet	$\frac{1}{2}$ x 18 in. x 9 ft.	"
1	"	$\frac{1}{2}$ x 14 in. x 7 ft.	"
1	"	$\frac{1}{2}$ x 12 in. x 8 ft.	"
1	"	$\frac{1}{2}$ x 14 in. x 6 ft.	"
1	"	10 gage x 24 in. x 8 ft.	"
4	Gage cock	$\frac{1}{2}$ in.	Brass
3	Solder-joint coupling	$\frac{1}{2}$ in. copper to $\frac{1}{2}$ in. inside I.P.S.	"
2	Solder-joint coupling	$\frac{1}{2}$ in. copper to $\frac{1}{2}$ in. outside I.P.S.	"
3	Solder-joint coupling	$\frac{1}{2}$ in. copper to $\frac{1}{2}$ in. inside I.P.S.	"
2	Solder-joint return bend	$\frac{1}{2}$ in. copper to copper	"
3	Sight glass	7 in. diam. x $\frac{1}{2}$ in. thick	Pyrex, annular edge
3	" "	4 in. diam. x $\frac{1}{2}$ thick	" " "
3	Steam trap	$\frac{1}{2}$ in.	Cast iron
2	Pump; Blackmer D.C.-10	10 g.p.m.	Bronze-fitted
1	Pump; Blackmer D.C.-5	5 g.p.m.	" - "
2	Electric motor	$\frac{1}{2}$ H.P.	—
1	" "	$\frac{1}{2}$ H.P.	—
5 sq. ft.	Wire cloth	100 mesh	Stainless steel
24	Bolts and nuts	$\frac{1}{2}$ in.-13 NC x 1 $\frac{1}{2}$ in. long	Brass
86	" " "	$\frac{1}{2}$ in.-13 NC x 1 $\frac{1}{2}$ in. long	Steel
78	" " "	$\frac{3}{8}$ in.-16 NC x 1 $\frac{1}{2}$ in. long	"
12	" " "	$\frac{1}{2}$ in.-20 NC x 2 $\frac{1}{2}$ in. long	"
1 sheet	Gasket material	48 x 48 x $\frac{1}{8}$ in.	Cranite

local company; those for the *University of New Hampshire* evaporator were made by the department. This was done in the following manner:

1. A sheet of 10-gage steel was laid out and cut with an acetylene torch into six sections, each of which had the dimensions of half a cone.

2. A simple furnace was made by stacking up fire-brick, laying a section of the cut steel sheet across the top, and applying the flame of a gas-fired blast lamp to the underside of the sheet.

3. When the sheet became red hot, it was removed and forged into shape by hand; about 2 or 3 reheatings were necessary before a forging was finished.

4. The cones were completed by welding the halves together, welding 3-inch steel flanges to the large ends, 2-inch nipples to the small ends, 1½-inch nipples for the condensate outlet pipes, and 1-inch nipples for the feed inlet.

Dished Heads. The dished heads for the evaporator bodies, the salt catchers, and the condensate receivers (Figure 6) were purchased, although they can be forged in a manner similar to that used in making the conical bottoms. This is most easily done by heating a steel sheet, placing it over the end of a piece of steel pipe of proper size, forging it to the desired shape, and then cutting it round with a torch.

Baskets. The design of the baskets is shown in Figure 4. In making the baskets, 2-foot lengths of 6-inch type O copper tubing were drilled for the steam inlets, condensate outlets, and vents, and then the tube sheets were cut from 48-ounce copper sheets, drilled with a ¾-inch drill, and silver-soldered to the ends of the 6-inch tubing. The ¾-inch heating tubes were cut long enough to project about ¼-inch beyond the tube sheets, to which they were then silver-soldered.

The baskets were finished by silver-soldering: (1) half of a 2-inch I.P.S. brass coupling to the steam inlet opening; (2) half of a ¾-inch copper solder-joint return bend to the condensate outlet opening; (3) a length of ¼-inch copper tubing, and a ¼-inch copper to ¼-inch inside I.P.S. coupling to the vent opening; (4) eight copper spacer fins to the outside of the basket to center it in the body; (5) a baffle hammered from a copper sheet and supported on four ¼-inch brass rods.

Liquor Samplers. To facilitate the testing of the concentration of the liquor being evaporated, each evaporator body was provided with a liquor sampler, made from a section of 2-inch pipe sufficiently long to permit the insertion of a hydrometer.

Entrainment Separators. Although the baffles above each basket are very effective in reducing entrainment, further separation is effected by means of cyclone-type separators, constructed in the manner shown in

Figure 3. The baffles in the separators were made from sheet copper and brazed in place. All other parts of the separators were fabricated by welding.

Condenser. The condenser, shown in Figure 5, was made from 8-inch standard steel pipe, and $\frac{3}{4}$ -inch hard copper tubing silver-soldered into copper tube sheets. The floating head is a section of 8-inch copper tubing.

Assembly. In erecting the evaporator, the supporting framework was constructed as shown in Figure 2, and the evaporator was assembled and placed as follows:

1. The body shells were elevated, placed in position, and bolted to the framework.

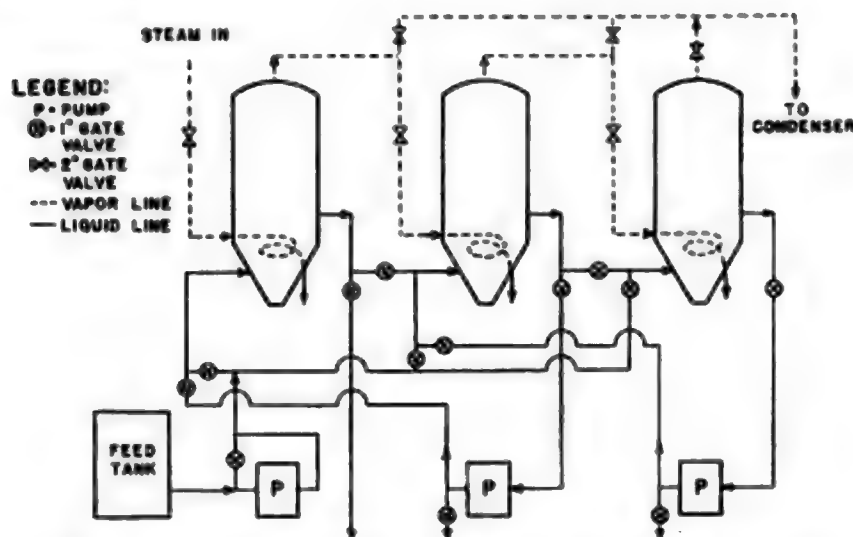


Figure 7. Piping Diagram for Triple-Effect Evaporator

2. The baskets were inserted from the bottom of the bodies, and brass nipples for the steam and vapor inlets and the vents were screwed tightly in place. Connections between the nipples and the evaporator bodies were made tight by the packing arrangement shown in Figures 3 and 4.

3. A length of $\frac{3}{4}$ -inch copper tubing was soldered into the soldered-joint fitting for the condensate outlet, and the conical bottom was bolted into position, using a *Cranite* gasket between the flanges.

4. The dished heads were then bolted to the tops of the bodies, the entrainment separators were connected, the condenser was set in place, and the vapor lines were installed.

5. The steam traps, coolers, condensate receivers, salt catchers, feed pumps, and water meter were then connected into position, and the piping was completed according to the diagram shown in Figure 7.

Operation. In making experimental runs with this evaporator, the valves in the feed and vapor lines are set for the desired conditions of operation, liquor is run into each body until it is about half way on the gage glass, steam is introduced to the first effect, the vacuum pump is turned on, and cooling water is passed through the condenser and coolers.

After the liquor begins to boil, the rate of feed is so adjusted as to maintain a constant level in each effect: the liquor should completely cover the basket, but should not be so high as to reach the baffles. When the proper level is determined, a mark is made on the gage glass as an aid in maintaining the proper level.

In collecting the condensate from each of the heating units and the condenser, the valves are adjusted so that one of each pair of receivers is hooked up at a time; it is also necessary to evacuate those receivers where the pressure in the heating units is below atmospheric. When one receiver is filled, the condensate is transferred to a second receiver which has previously been evacuated if necessary.

TYPICAL EXPERIMENT

Object. To determine the economy and the overall heat-transfer coefficients for the evaporation of water in a triple-effect evaporator operated with forward feed.

Procedure. 1. Pump water from the feed tank through the water meter into the first effect and let it flow into the second and third effects. When it reaches the proper levels, close the feed valve and the liquor valves between the effects.

2. Turn on the steam to the first effect and adjust the pressure to the desired value.

3. Adjust the valves of the condensate receivers so that the left-hand receiver of each pair is open to its respective condensate cooler, and then turn on the vacuum pump and evacuate the last effect and the necessary condensate receivers.

4. As evaporation proceeds, adjust the feed valve and the liquor valves between the effects so as to maintain the liquor levels at the marks on the water gages.

5. Read all thermometers and pressure gages occasionally until they indicate that conditions have become substantially constant; this will probably require about 2 hours. As a condensate receiver becomes filled, shift the flow of condensate to its mate, and drain the water which has been collected. When emptying a receiver, first read the level, and then drain as far as possible without running the level below the lowest gage mark.

6. After conditions have become constant, start a timed run. Read

the water meter and levels in all receivers, and then record the following data at intervals of several minutes for about 2 hours:

- a. Pressure of steam to first effect.
- b. Pressure or vacuum in each effect.
- c. Temperatures of liquid and vapor in each effect.
- d. Temperature of feed.

Read the water meter several times during the run, and record the quantity of condensate drained from each receiver.

At the end of the timed run, take a final reading of the water meter, and record the levels in the condensate receivers.

7. Calculate:

- a. The radiation loss from each effect.
- b. The pounds of water evaporated per pound of steam.
- c. The overall heat transfer coefficient of each effect.

Experimental Data and Calculations. The experimental results are given in Tables 2 and 3, and the pressures and liquid and vapor temperatures during the course of the timed run are shown graphically in Figure 8. The average of each set of readings was used in subsequent calculations.

It is evident that the liquor temperature in each effect was a few degrees higher than the corresponding vapor temperature. This was due to the fact that the liquor thermometer was under a hydrostatic head, for this thermometer was placed about halfway between the top and the bottom of the basket in order to obtain the average liquor temperature.

As an aid in making the calculations, a diagrammatic sketch, Figure 9, was prepared to show the various pressures, temperatures, and quantities of feed and condensate involved.

Heat Balances. The heat balances for the three effects, taking the temperature of the liquid in the effect as the datum temperature, are as follows:*

First effect

$$W(L.H._{239.0^{\circ}\text{F.}}) = X[L.H._{239.0^{\circ}\text{F.}} + (236.0 - 239.0)(0.5)] - F(63.9 - 239.0)(1.0) + R_1$$

Second effect

$$X(L.H._{236.0^{\circ}\text{F.}}) = Y[L.H._{211.7^{\circ}\text{F.}} + (209.5 - 211.7)(0.5)] - (F - X)(239.0 - 211.7)(1.0) + R_2$$

* In each of the heat balances, a term is included to account for the difference in enthalpy of the vapor between the datum and vapor temperatures. However, this term is very small and, for practical purposes, may be neglected.

TABLE 2

Experimental Results for Evaporation of Water in a Triple-Effect Evaporator

Time	Gage readings				Liquor temperatures, °F.			Vapor temperatures, °F.			Feed	
	Steam, lb./ in. ²	1st effect, lb./in. ² pres- sure	2nd effect, in. Hg vacu- um	3rd effect, in. Hg vacu- um	1st effect	2nd effect	3rd effect	1st effect	2nd effect	3rd effect	Meter reading, ft. ³	Temp. °F.
10:42 A.M.	18.5	7.3	1.0	17.1	236	212	177	234	210	175	96.37	65.3
10:50	18.8	7.7	0.8	16.0	240	212	180	237	210	177	97.65	64.4
10:57	15.5	3.7	3.4	16.0	—	—	—	—	—	—	—	—
11:00	—	—	—	—	222	204	176	220	202	174	98.60	64.4
11:02	14.5	1.6	4.0	16.2	226	208	178	222	204	176	—	—
11:04	15.5	4.3	3.7	16.2	230	208	180	226	206	174	99.39	64.4
11:10	17.5	5.5	0.0	16.4	233	212	179	230	210	176	99.85	64.4
11:12	19.0	7.5	1.0	16.1	238	212	180	237	210	177	—	—
11:14	20.0	7.2	1.0	15.8	236	211	180	234	210	177	—	—
11:16	20.0	7.5	0.0	16.0	238	213	180	234	210	177	—	—
11:17	20.5	8.4	1.0	14.8	240	213	184	236	210	180	—	—
11:22	19.0	7.0	1.0	16.0	238	213	180	235	211	177	101.00	64.4
11:24	21.0	9.2	0.8	16.0	242	212	180	239	210	178	—	—
11:26	22.5	10.6	0.8	16.0	245	212	180	242	210	178	—	—
11:27	21.5	9.7	0.0	16.5	242	213	181	240	210	179	—	—
11:37	20.5	9.0	0.0	16.0	242	212	180	240	210	178	102.88	64.4
11:40	22.0	10.0	0.0	16.0	—	—	—	—	—	—	—	—
11:43	19.0	6.4	0.0	15.5	234	212	180	230	210	178	—	—
11:45	18.6	6.6	1.0	15.5	236	211	182	232	210	178	104.00	63.5
11:48	20.0	7.9	0.0	15.2	240	213	182	236	210	179	—	—
11:50	21.7	9.6	0.0	15.2	242	213	182	240	210	180	104.44	63.5
11:53	21.3	9.4	0.0	15.2	242	213	182	240	210	180	—	—
11:55	21.0	9.2	0.0	15.2	242	213	182	238	210	180	—	—
11:58	21.2	9.3	0.0	15.4	242	212	180	239	210	179	—	—
12:04 P.M.	21.0	9.0	0.0	15.1	240	212	182	238	210	180	104.95	63.5
12:10	22.0	8.5	0.0	15.0	240	212	182	236	210	180	106.15	62.6
12:12	21.8	7.9	0.0	16.1	239	212	178	235	210	176	—	—
12:17	21.0	9.0	1.0	16.0	241	213	180	238	210	178	106.82	62.6
12:20	22.5	10.0	0.8	16.0	243	212	180	240	210	178	106.99	63.5
12:25	22.0	9.3	0.8	15.5	241	212	181	238	210	179	—	—
12:30	22.4	9.0	1.0	15.3	240	212	180	238	210	178	107.92	63.5
12:31	23.0	9.7	0.7	15.0	242	212	181	240	210	179	—	—
12:38	21.5	9.0	0.8	15.5	242	212	180	239	210	178	—	—
12:40	23.0	9.7	1.0	15.5	242	212	180	240	210	178	108.75	63.5
12:42	23.8	11.0	0.6	15.2	246	212	180	243	210	178	108.92	63.5
Average...	20.4	8.1	0.8	15.7	239.0	211.7	180.0	236.0	209.5	177.8	—	63.9

TABLE 3
Levels in Condensate Receivers
(Height in inches)

	1st Effect		2nd Effect		3rd Effect		Condenser	
Receiver No.	1	2	3	4	5	6	7	8
Calibration, lb./in.	2.93	2.92	2.94	2.92	2.92	2.91	5.20	5.16
Time								
10:42 A. M.		2.55	6.15		13.40			3.6
10:50					19.80*	0.95		
10:53	0.00	19.98*						
10:56							2.02	10.35*
10:58			19.85*	1.2				
11:02	19.59*	0.37						
11:13					1.45	19.11*		
11:17							10.81*	1.40
11:20			1.17	18.65*				
11:23	1.30	19.85*						
11:35					18.24*	1.97		
11:37	19.30*	0.60						
11:40							1.45	10.44*
11:47			19.46*	1.28				
11:52	1.55	19.59*						
11:59					1.77	19.65*		
12:01 P. M.							10.58*	1.27
12:05	19.00*	1.21						
12:08			1.3	18.92*				
12:20	0.63	19.43*						
12:24					19.07*	1.31		
12:25							1.36	10.20*
12:30			18.64*	1.65				
12:34	19.26*	1.58						
12:42		9.30*		9.10*		13.28*	7.20*	
Total in.	73.67	81.84	49.33	42.54	40.49	47.81	23.76	24.72
Lb. per receiver. . .	215.8	238.9	145.0	124.3	118.3	139.3	123.2	127.7
Lb. per effect.	454.7		269.3		257.6		250.9	

* Levels before draining; other figures are the levels after draining.

Third effect

$$Y(L.H._{209.6^{\circ}\text{F.}}) = Z[L.H._{180.0^{\circ}\text{F.}} + (177.8 - 180.0)(0.5)] \\ - (F - X - Y)(211.7 - 180.0)(1.0) + R_3$$

In these equations, 0.5 and 1.0 are the heat capacities of water vapor and liquid water, respectively; the other numerical terms are temperatures shown in Figure 9; and the symbols have the following significance:

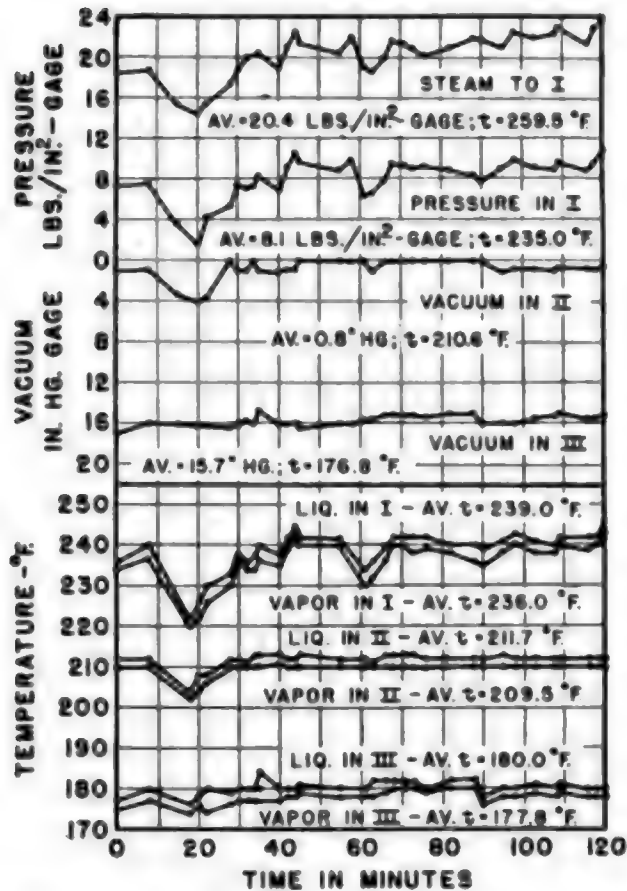


Figure 8. Temperature and Pressure Logs of Evaporation Experiment

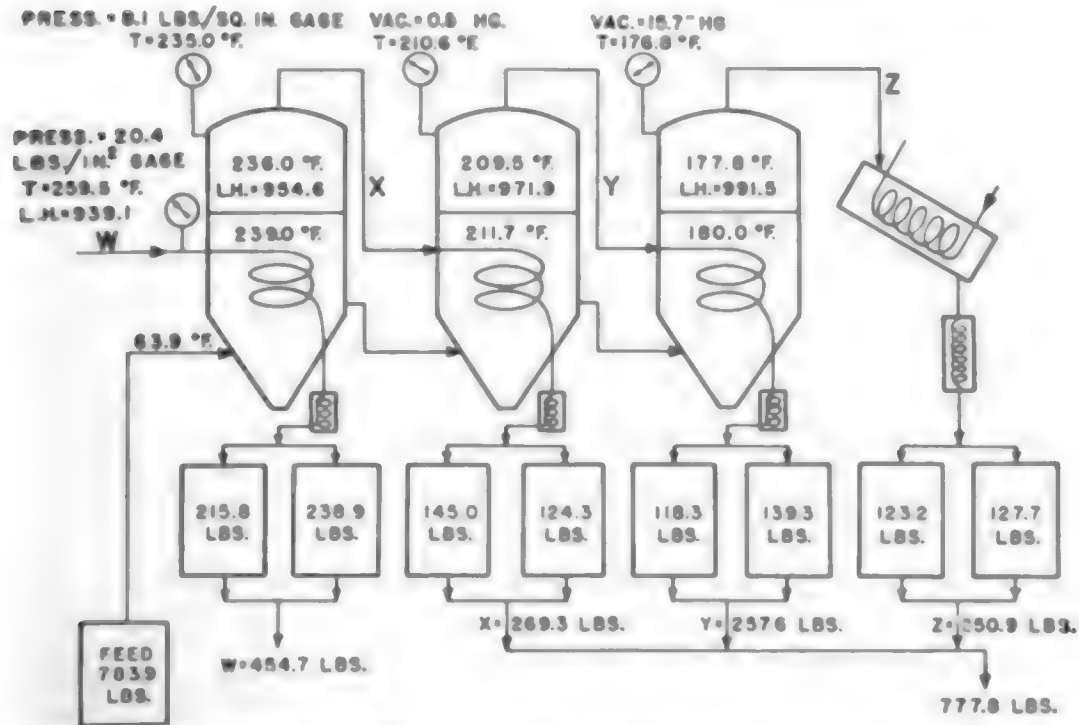


Figure 9. Average Results of Evaporation Experiment

F = weight of feed, lb./2 hr.

W = weight of steam condensed, lb./2 hr.

X, Y, Z = weight of water evaporated in 1st, 2nd, and 3rd effect, respectively, lb./2 hr.

R_1, R_2, R_3 = radiation loss from 1st, 2nd, and 3rd effects, respectively, B.t.u./2 hr.

$L.H.$ = latent heat at temperature indicated, B.t.u./lb.

The calculations for the 1st effect are given below; the calculations for the other two effects are similar and are not included here. However, the calculated results for all of the effects are summarized in Table 4.

TABLE 4
Summary of Calculated Results for Evaporation of Water in a Triple-Effect Evaporator

Item No.	Item	Effect No.		
		1	2	3
1	Time of run, hr.....	2	2	2
2	Datum temperature, °F.....	239	211.7	180.0
3	Heat from steam or vapor, B.t.u....	427,000	260,000	250,000
4	Heat in vapor, B.t.u.....	256,000	249,700	248,500
5	Heat in feed, B.t.u.....	-137,200	14,050	8,150
6	Radiation loss, B.t.u.....	33,800	24,350	9,650
7	Radiation loss, per cent.....	7.9	8.9	3.9
8	Area of heating surface, ft. ²	11.29	11.29	11.29
9	Overall temperature drop, °F.....	20.5	24.3	29.5
10	Overall heat-transfer coefficient, B.t.u./(hr. × ft. ² × °F.).....	922	475	376
11	Economy, lb. of water evaporated per lb. of steam.....	1.71		

Item 3: Heat given up by steam, B.t.u./ 2 hr.

$$= W(L.H._{259.5^{\circ}\text{F.}}) = 454.7 \times 939.1 = 427,000$$

where

454.7 = weight of steam condensed, lb./2 hr.

939.1 = latent heat of vaporization of water at 259.5°F..
B.t.u./lb.

Item 4: Heat in vapor leaving effect, B.t.u./hr.

$$\begin{aligned}
 &= X[L.H._{239.0^{\circ}\text{F.}} + (236.0 - 239.0)(0.5)] \\
 &= 269.3 [952.9 + (236.0 - 239.0)(0.5)] \\
 &= 269.3 (952.9 - 1.5) = 256,000
 \end{aligned}$$

where

269.3 = weight of vapor condensed, lb./2 hr.

952.9 = latent heat of evaporation of water at 239°F.,
B.t.u./lb.

236.0 = temp. of vapor, °F.

239.0 = temp. of liquor, °F.

0.5 = heat capacity of water vapor, B.t.u./(lb. × °F.).

Item 5: Heat in feed, B.t.u./2 hr.

$$= 783.9 (63.9 - 239.0) \times 1.0 = -137,200$$

where

783.9 = weight of water fed to 1st effect, lb./2 hr.

63.9 = average temperature of feed, °F.

239.0 = datum temperature = liquor temperature, °F.

1.0 = heat capacity of liquid water, B.t.u./(lb. × °F.).

Item 6: *Radiation loss, B.t.u./2 hr.*

$$= \text{Item 3} - (\text{Item 4} - \text{Item 5})$$

$$= 427,000 - (256,000 + 137,200)$$

$$= 33,800$$

Item 7: *Radiation loss, per cent*

$$= \frac{\text{Item 6}}{\text{Item 3}} \times 100 = \frac{33,800}{427,000} \times 100 = 7.9$$

Overall Heat-Transfer Coefficients. The overall heat-transfer coefficient for each effect was calculated by means of the equation:

$$U = \frac{Q/\theta}{A\Delta t}$$

where

U = overall heat-transfer coefficient, B.t.u./(hr. × ft.² × °F.).

Q = heat transferred through heating surface, B.t.u. = Item 3.

θ = time of run, hr. = Item 1.

Δt = temperature difference between condensing steam and boiling liquor, °F. = Item 9.

A = mean area of heating surface, ft.² = Item 8.

Each basket has the following dimensions

No. of tubes = 18

Outside diameter of tubes, in. = 0.875

Wall thickness of tubes, in. = 0.032

Length of basket, ft. = 2

Outside diameter of basket, in. = 6.0

Wall thickness of basket, in. = 0.072

Therefore,

$$1. \text{ Area of tubes} = \frac{0.875 - 0.032}{12} (\pi)(2)(18) = 7.955 \text{ ft.}^2$$

$$2. \text{ Area of basket surface} = \frac{6.00 - 0.072}{12} (\pi)(2) = 3.105 \text{ ft.}^2$$

$$3. \text{ Area of basket ends} = (2)(\pi) \left[\frac{(6.00 - 0.072)^2}{4 \times 144} - 18 \frac{(0.875)^2}{4 \times 144} \right] \\ = 0.234 \text{ ft.}^2$$

$$\text{and the total mean area} = 7.955 + 3.105 + 0.234 \\ = 11.294 \text{ ft.}^2 = \text{Item 8.}$$

For the 1st effect, the calculation of the overall coefficient is

$$U_1 = \frac{Q/\theta}{A\Delta t} = \frac{\text{Item 3}/\text{Item 1}}{\text{Item 8} \times \text{Item 9}} = \frac{427,000/2}{(11.29)(20.5)} \\ = 922 \text{ B.t.u.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}) = \text{Item 10.}$$

Economy of Evaporation

= lb. water evaporated per lb. steam

$$= \frac{777.8}{454.7} = 1.71 = \text{Item 11.}$$

where

777.8 = weight of water evaporated, lb.

454.7 = weight of steam used, lb.

Comments on Results. This experiment shows that: (1) the overall coefficients are very high and, as expected, they decrease as the temperature decreases; and (2) the economy of evaporation is much lower than is usually associated with triple-effect operation. This was due, mainly, to the fact that the feed was very cold, and large quantities of steam were required to heat it to the boiling temperature. Furthermore, the experiment was performed with no insulation on the evaporator, and radiation losses were higher than they would otherwise have been. Nevertheless, the radiation losses were very low compared to the total heat transferred.

NOMENCLATURE

A = area of heating surface, ft.²

F = weight of feed, lb./2 hr.

$L.H.$ = latent heat, B.t.u./lb.

Q = heat transferred, B.t.u.

R_1, R_2, R_3 = radiation loss from 1st, 2nd and 3rd effects, respectively,
B.t.u./2 hr.

Δt = temperature difference, °F.

U = overall heat-transfer coefficient, B.t.u./(hr. \times ft.² \times °F.).

W = weight of steam condensed, lb./2 hr.

X, Y, Z = weight of water evaporated in the 1st, 2nd, and 3rd
effects, respectively, lb./2 hr.

θ = time, hr.

A SINGLE-EFFECT LONG-TUBE EVAPORATOR

Designed and Constructed by
The Department of Chemical Engineering
University of North Dakota

Description. This evaporator is constructed of standard pipe and fittings and brass condenser tubes; it has an overall height of approximately 13 feet. The heating section is made from a length of 8-inch steel pipe, and has 12 tubes, $\frac{1}{2}$ inch in diameter and 8 feet long, which provide a total heating surface of 17.3 square feet. Effective separation of vapor from entrained liquid is provided by an entrainment separator made from a 4-foot length of 12-inch pipe.

A line drawing of the evaporator and details of the tube-sheet assembly are shown in Figure 1.

Materials and Cost. The materials used in constructing the evaporator are given in Table 1; this does not include materials for the condenser. The cost of the items listed in the table is approximately \$220.

Notes on Construction. The tubes are secured by packing plates and conical rubber gaskets.¹ The tube sheets and packing plates are made of $\frac{1}{2}$ -inch steel plate, and each set is drilled to receive four $\frac{1}{2}$ -inch bolts and twelve $\frac{1}{2}$ -inch tubes, as shown in the drawing. The tube holes are drilled slightly larger than the outside diameter of the tubes, and are countersunk to a depth of $\frac{3}{4}$ inch with a drill $\frac{1}{8}$ inch smaller in diameter than the outside diameter of the gasket. The bolts are made secure by welding their heads to the inside of the tube sheets.

The tubes are cut so that they project about 1 inch beyond the tube sheet on either end, and they are then assembled in the flanged section of the 8-inch pipe. Rubber gaskets are slipped over the ends of each tube and forced into place by the packing plates.

Comments on Design. 1. The method used for securing the tubes has proved to be very satisfactory: no leaks have been experienced at pressures as high as 60 pounds per square inch gage. The particular value of this method, however, is in the ease with which tubes can be replaced. This makes it possible to change the capacity of the evaporator by varying the number of tubes in the bundle.

2. The actual dimensions of the evaporator are of no great importance; the height and capacity can be varied depending upon the head room available and upon individual preference.

3. At the *University of North Dakota*, a surface condenser is used with

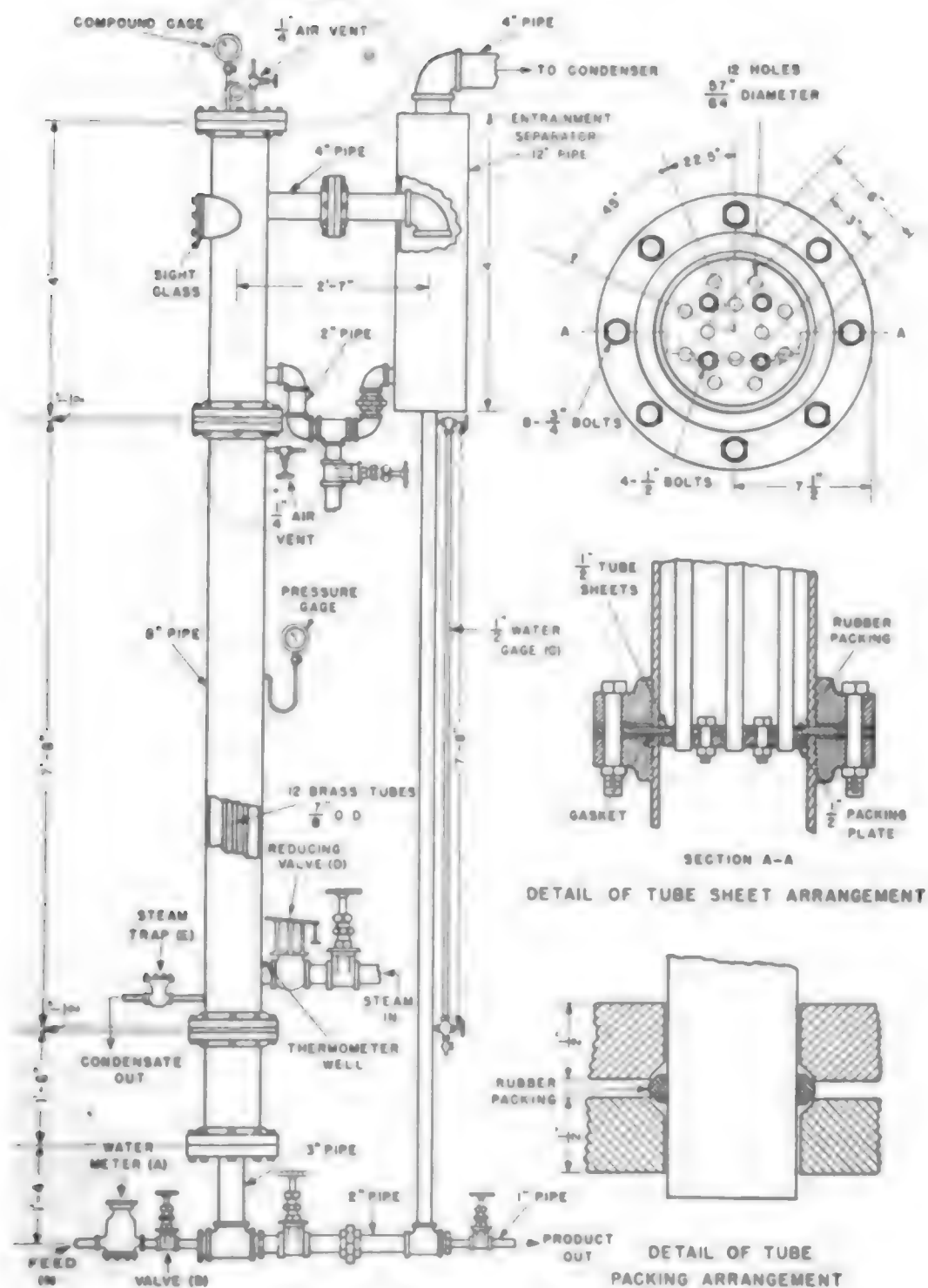


Figure 1. Long-Tube Evaporator—University of North Dakota

TABLE 1
Bill of Materials for Single-Effect Long-Tube Evaporator
 University of North Dakota

Quantity	Description	Size	Material
1 piece	Std. pipe	12 in. x 4 ft. long	Black wrought steel
1 piece	" "	8 in. x 7 ft. 8 in. long	" " "
1 piece	" "	8 in. x 4 ft. long	" " "
1 piece	" "	8 in. x 1½ ft. "	" " "
1 piece	" "	4 in. x 1½ ft. "	" " "
2 ft.	" "	3 in.	" " "
12½ ft.	" "	2 in.	" " "
3	Std. nipple	4 in. x 12 in. long	" " "
2	" "	4 in. x 6 in. "	" " "
1	" "	2 in. x 12 in. "	" " "
1	Std. tee	2 in.	" " "
1	" "	3 x 2 x 3 in.	" " "
1	" "	2 x 1 x 2 in.	" " "
2	Std. 90° elbow	4 in.	" " "
4	" " "	2 in.	" " "
1	Ground-joint union	2 in.	" " "
1	Bushing	3 x 1 in.	" " "
2	Std. flange union	8 in.	Cast iron
2	" " "	4 in.	" "
2	Companion flange	8 in. x 13½ in. O.D.	" "
1	" "	3 in. x 13½ in. O.D.	" "
1	Blind flange	13½ in. O.D.	" "
2	Std. gate valve	2 in.	Brass
4	" " "	1 in.	"
2	" " "	½ in.	"
1	Steam trap	½ in., 2000 lb./hr. capacity	Cast iron
1	Pressure gage	0 to 100 lb.; 3½ in. face	Brass
1	Compound gage	0 to 30 in. vac., 0 to 15 lb. press.; 3½ in. face	"
1	Water gage	½ in.	"
1	Water meter	½ in.	Bronze
1	Pressure-reducing valve	2 in.	Steel; bronze trimmed
12	Std. condenser tube	½ in. O.D. x 16 gage x 8 ft. long	Brass
2	Round plate	13 in. diam. x ½ in. thick	Steel
2	" "	12 in. diam. x ½ in. thick	"

TABLE 1—(Concluded)

Quantity	Description	Size	Material
2	Round plate	8 in. diam. x $\frac{1}{2}$ in. thick	Steel
1	Sight glass	6 in.	Pyrex glass
6	Gasket	8 in. diam. x $\frac{1}{8}$ in. thick	Cranite
4	"	4 in. diam. x $\frac{1}{8}$ in. thick	"
24	Conical gasket	$\frac{11}{16}$ in. I.D. x $\frac{11}{16}$ in. O.D. x $\frac{1}{4}$ in. thick	Rubber

this evaporator so that the condensate can be collected and weighed. However, if there is no need to collect the condensate, a contact condenser is preferable because of its lower cost.

4. Regardless of the type of condenser used, it is desirable to have an ejector or vacuum pump to permit operation under reduced pressure.

TYPICAL EXPERIMENT

Object. To study the effects of temperature drop and liquor level on the capacity and the overall heat-transfer coefficient of the evaporator.

Operation. Feed, measured by water meter *A*, is introduced into the evaporator through valve *B* to a predetermined level on water gage *C*. Steam is introduced into the steam chest, and pressure-reducing valve *D* is set for the desired pressure. The steam condensate from trap *E* is passed through a cooler and collected. As evaporation proceeds, feed is introduced continuously at a rate sufficient to maintain the desired level in the evaporator. The evaporator is run for approximately 30 minutes to attain equilibrium conditions, and then data are taken over a 10 to 15 minute interval, during which time the following measurements are recorded:

1. Quantity and temperature of water introduced.
2. Temperatures and pressures of the steam and vapor.
3. Weight of condensate from condenser.
4. Weight of steam condensate from trap.

The experiment is repeated at various steam pressures and liquid levels.

A separate series of runs is made to determine the radiation losses from the steam chest. During these runs, the evaporator is operated without introducing feed, and the heat losses are calculated from the quantities of steam condensed at various pressures.

TABLE 2
Experimental Data and Calculated Results for Evaporation of Water in a Long-Tube Evaporator

Sym- bol	Item	Run number									
		1	2	3	4	5	6	7	8	9	10
θ_m	Time, min.	10	10	10	10	10	10	10	10	10	10
W_1	Steam used, lb.	41	41	65	61	91	92	116	123	136	158
t_1	Steam temp., °F.	223.5	223.7	228.2	228.2	231.8	231.8	234.5	234.5	236.3	238.1
λ_1	Latent heat of steam, B.t.u./lb.	963	963	960	960	958	958	956	956	955	954
W_2	Water evaporated, lb.	32	32	52	54	78	79	106	111	126	147
t_2	Temp. of vapor, °F.	212	212	212	212	212	212	212	212	212	212
λ_2	Latent heat of vapor, B.t.u./lb.	970	970	970	970	970	970	970	970	970	970
t_3	Temp. of feed, °F.	145	145	136	136	140	150	158	165	164	170
q_1	Heat supplied by steam, B.t.u.	39,500	39,500	62,400	58,600	87,200	88,100	110,800	118,800	130,000	150,750
q_2	Radiation loss from steam chest, B.t.u.	1,180	1,180	1,240	1,240	1,280	1,280	1,340	1,340	1,355	1,365
Q	Heat through heating surface, B.t.u.	38,320	38,320	61,160	57,360	85,920	86,820	109,460	116,460	128,645	149,365
Δt	Apparent temp. drop, °F.	11.5	11.7	16.2	16.2	19.8	19.8	22.5	22.5	24.3	26.1
U	Apparent overall coefficient, B.t.u./ (hr. \times ft. ² \times °F.)	1,156	1,134	1,310	1,220	1,506	1,522	1,657	1,798	1,837	1,985
q_3	Heat in vapor above feed temp., B.t.u.	33,200	33,300	54,400	56,600	81,300	81,500	107,500	112,900	128,200	148,600
q_4	Total heat lost by radiation, B.t.u.	6,300	6,200	8,000	2,100	5,900	6,600	3,300	5,900	1,900	2,150
H/L	Heat loss, per cent	18.6	15.7	12.8	3.6	6.8	7.5	3.0	5.0	1.4	1.4
E	Lb. water evaporated per lb steam used	0.761	0.761	0.800	0.865	0.858	0.960	0.905	0.904	0.928	0.931

Typical Data. The experimental data and calculated results for ten runs are given in Table 2, and the effect of temperature drop on the overall coefficient is shown graphically in Figure 2.

Calculations for Run No. 1.

$$\begin{aligned}
 q_1 &= \text{heat supplied by steam, B.t.u./10 min.} \\
 &= (\text{weight of steam used, lb.}) \times (\text{latent heat of steam, B.t.u./lb.}) \\
 &= W_1 \times \lambda_1 \\
 &= 41 \times 963 = 39,500
 \end{aligned}$$

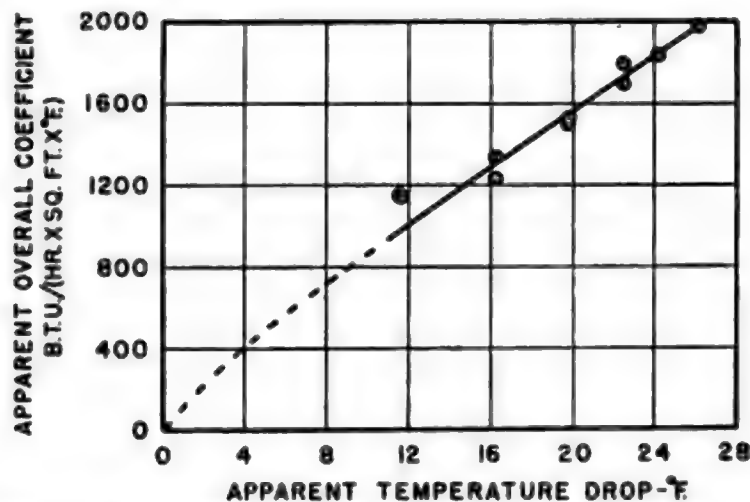


Figure 2. Effect of Temperature Drop on Overall Heat-Transfer Coefficient of a Long-Tube Evaporator

$$\begin{aligned}
 Q &= \text{heat transferred through heating surface, B.t.u./10 min.} \\
 &= (\text{heat supplied by steam, B.t.u.}) - (\text{heat lost by radiation, B.t.u.}) \\
 &= q_1 - q_2 \\
 &= 39,500 - 1,180 = 38,320
 \end{aligned}$$

$$\begin{aligned}
 \Delta t &= \text{apparent temperature drop, } ^\circ\text{F.} \\
 &= (\text{temp. of steam, } ^\circ\text{F.}) - (\text{temp. of vapor, } ^\circ\text{F.}) \\
 &= t_1 - t_2 \\
 &= 223.5 - 212 = 11.5
 \end{aligned}$$

$$\begin{aligned}
 U &= \text{apparent overall coefficient of heat transfer, B.t.u./} \\
 &\quad (\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}) \\
 &= \frac{Q/\theta}{A\Delta t} = \frac{38,320/0.166}{17.3 \times 11.5} = 1,156
 \end{aligned}$$

where

$$17.3 = \text{area of heating surface, ft.}^2 = A.$$

$$0.166 = \text{time of run, hr.} = \frac{\theta_m}{60} = \frac{10}{60} = \theta.$$

$$\begin{aligned} q_3 &= \text{heat in vapor above feed temp., B.t.u./10 min.} \\ &= (\text{water evaporated, lb.}) \times [(\text{latent heat of evaporation, B.t.u./lb.}) + (\text{sensible heat, B.t.u.})] \\ &= W_2 \times [\lambda_2 + (t_2 - t_3) \times 1.0] \\ &= 32 \times [970 + (212 - 145) \times 1.0] = 33,200 \end{aligned}$$

$$\begin{aligned} q_4 &= \text{total heat lost by radiation, B.t.u./10 min.} \\ &= (\text{heat supplied by steam, B.t.u.}) - (\text{heat in vapor above feed temp., B.t.u.}) \\ &= q_1 - q_3 \\ &= 39,500 - 33,200 = 6,300 \end{aligned}$$

H.L. = heat loss, per cent

$$= \frac{\text{total heat lost by radiation}}{\text{total heat supplied by steam}} \times 100$$

$$= \frac{q_4}{q_1} \times 100$$

$$= \frac{6,300}{39,500} \times 100 = 15.6$$

E = lb. water evaporated per lb. steam used

$$= \frac{W_2}{W_1} = \frac{32}{41} = 0.781$$

NOMENCLATURE

A = area of heating surface, ft.²

E = lb. water evaporated per lb. of steam.

H.L. = heat loss, per cent.

Q = heat transferred through heating surface, B.t.u.

*q*₁ = heat supplied by steam, B.t.u.

*q*₂ = radiation loss from steam chest, B.t.u.

*q*₃ = heat in vapor above feed temp., B.t.u.

*q*₄ = total heat lost by radiation, B.t.u.

*t*₁ = temp. of steam, °F.

*t*₂ = temp. of vapor, °F.

*t*₃ = temp. of feed, °F.

Δt = apparent temp. drop, °F.

U = apparent overall heat transfer coefficient, B.t.u./
(hr. \times ft.² \times °F.).

W_1 = steam used, lb.

W_2 = water evaporated, lb.

θ = time of run, hr.

θ_m = time of run, min.

λ_1 = latent heat of the steam, B.t.u./lb.

λ_2 = latent heat of the vapor, B.t.u./lb.

REFERENCE

1. BADGER AND McCABE, "Elements of Chemical Engineering," McGraw-Hill Book Company, New York, 1936, 2nd ed., pp. 175-7.

A SALTING-TYPE EVAPORATOR

Designed by

The Department of Chemical Engineering
Rose Polytechnic Institute*

Description. This evaporator, which is representative in all details of a full-sized, salting-type evaporator, consists of a vapor head, a calandria, a cone bottom, a salt settler, and a centrifugal type of catchall. The calandria is designed for working pressures up to 50 pounds per square inch, and the other parts of the unit are designed for full vacuum. Industrial-type thermometers and calibrated pressure gages are provided for temperature and pressure readings.

Figure 1 shows the assembled evaporator, and Figures 2 and 3 show the details of construction.

Notes on Construction. 1. The apparatus was fabricated of mild steel. The work of rolling the cylindrical and conical portions of the shell from plate, the flanging and dishing of the heads, the assembly of the calandria, and all welding were done by a local boiler works, which had never before built any process equipment. A complete set of assembly and detail drawings was furnished, together with a verbal description of the apparatus and its operation. No trouble was encountered during fabrication. Any small boiler works which has rolls for forming steel plate into cylinders, and are welding equipment could build a similar evaporator.

2. The filter bottom in the salt settler consists of two steel plates, the lower one $\frac{1}{2}$ -inch thick and the upper one $\frac{3}{4}$ -inch thick, each of which is perforated with $\frac{3}{4}$ -inch diameter holes on 1-inch centers. Between these plates is a filter medium made by placing a piece of light-weight muslin between two circular pieces of 16-mesh bronze screen and fastening these together around the edges with a stapler.

3. All sight glasses are of *Pyrex*. Leaks are prevented by soft rubber gaskets, $\frac{1}{8}$ -inch thick, cemented in place on each side of the glass with *Permatex No. 2* cement.

4. The salt settler door has a $\frac{3}{4}$ -inch square asbestos gasket set in a groove. A little *Permatex No. 2* cement placed on the gasket before closing the door makes a tight joint without the need of excessive pressure on the handles.

Cost. The total cost of the materials and the fabrication labor of the completed evaporator was \$370; this did not include the cost of erection or the auxiliary equipment that is needed when making a run.

* Reported by O. S. Knight.

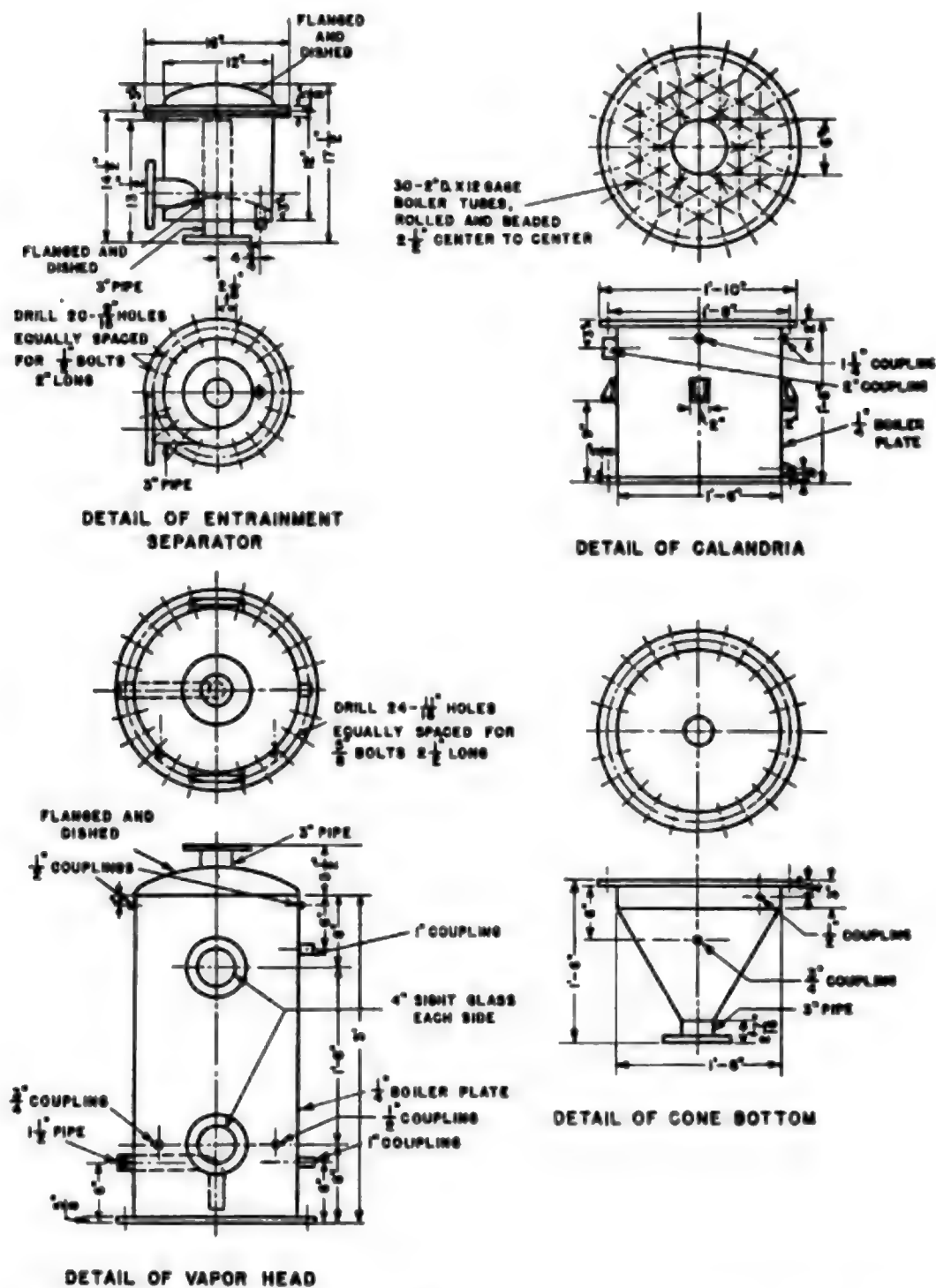


Figure 2. Details of Construction of Entrainment Separator, Vapor Head, Calandria, and Cone Bottom

Erection and pipe fitting were done by students as a part of their laboratory work.

Comments on Design. 1. Insulation would improve the performance of this evaporator, although results are quite satisfactory without it.

2. Because the evaporator is operated only intermittently, there is some darkening of the crystallized product due to iron compounds; the

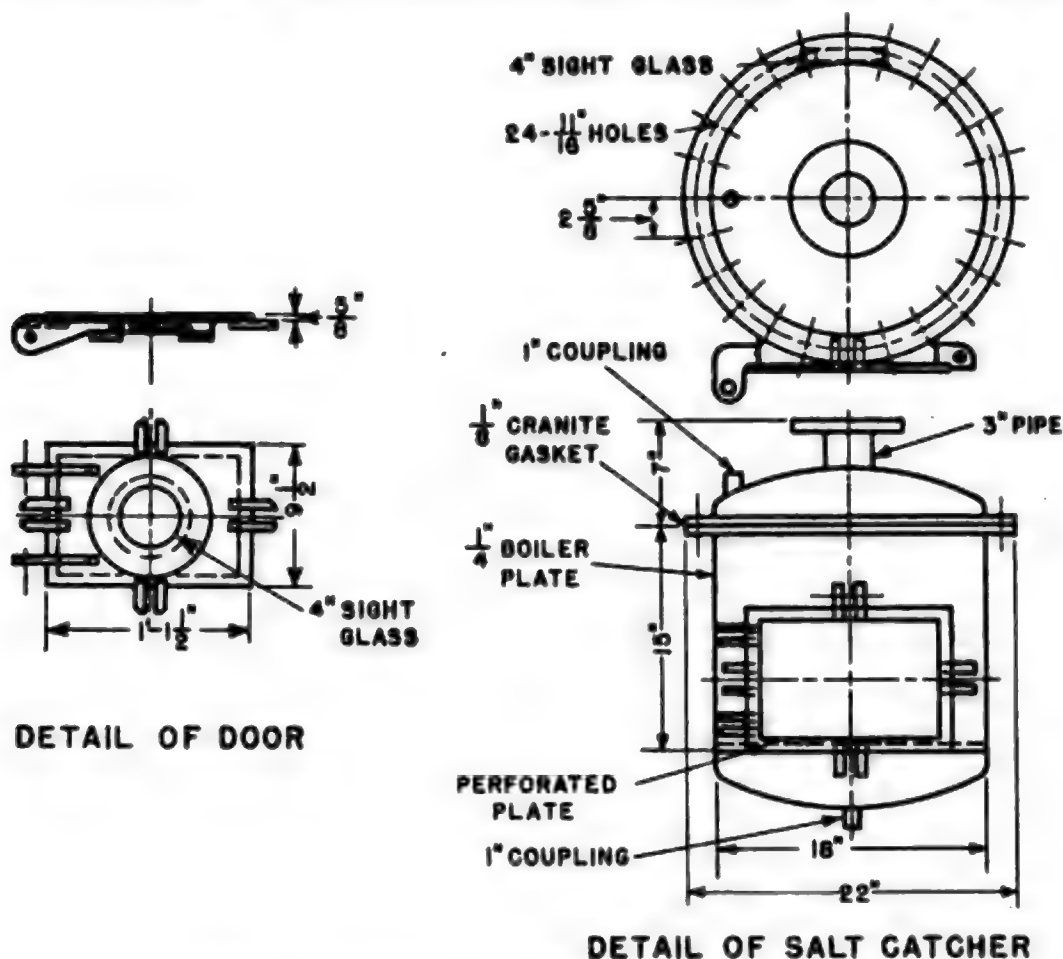


Figure 3. Details of Construction of Salt Catcher

use of *Monel* metal in the construction of those parts which come in contact with brine would overcome this difficulty, and is advisable where a pure product is desired.

Auxiliary Equipment. The following equipment, which is not shown in the drawings, is used when making a run with this evaporator under vacuum:

1. Surface condenser: 18 square feet, single pass. (A condenser with 30 to 40 square feet of heat-transfer surface would be better.)

2. Vacuum pump: Kinney VSD 778, 26.2-cubic feet per minute displacement; water cooled; driven by $1\frac{1}{2}$ -H.P. motor.

3. Feed pump, for use when working under positive pressure: Fairbanks-Morse H-600-42 shallow-well ejector pump; capacity 2 gallons per minute against a 92-foot head of water; driven by a direct-coupled $\frac{1}{4}$ -H.P. motor.

4. Water meter: Pittsburgh Equitable Meter Company's $\frac{1}{8}$ -inch IMO.

5. Tanks for collecting and measuring steam and vapor condensate: two of 75-gallon capacity, designed for vacuum service.

6. Tanks for mixing and holding feed: three 55-gallon oil drums with top heads removed.

7. Platform scales: capacity 500 pounds.

8. Steam regulating valve: Klipfel $1\frac{1}{2}$ inch.

9. Steam calorimeter.

10. Steam trap: Hoffman $\frac{1}{2}$ -inch radiator trap (Crane Company).

Operation. 1. With valve V_1 closed and valves V_2 and V_3 open, feed is introduced through valve V_4 until it fills the salt settler and rises in the evaporator to the top of the upper tube sheet; during a run, the feed is introduced at a rate sufficient to maintain this level.

2. When the evaporator is filled, steam at the desired pressure is admitted into the calandria through valve V_4 . Cooling water is passed through the condenser and the cooling jacket of the vacuum pump, the vacuum pump is started, and the vapor condensate receiver is placed under vacuum. The steam condensate receiver, however, is left open to the atmosphere, and vent cock C_1 is cracked to eliminate air from the calandria.

3. To remove salt from the settler, valve V_2 is closed, valve V_3 and cock C_2 are opened wide, and the solution in the salt settler is drawn up into the evaporator body. Valve V_3 is then closed, and the salt is removed through the door of the settler.

TYPICAL EXPERIMENT

Object. To determine the overall heat-transfer coefficient, the capacity, and the economy of the salting-type evaporator when evaporating saturated sodium chloride brine.

Procedure. 1. Prepare approximately 100 gallons of saturated sodium chloride brine in 55-gallon drums. Use the *Lightnin* mixer for agitation. Save a sample from each batch for determination of the sodium chloride content.

2. Study the evaporator to familiarize yourself with the equipment. Adjust the valves for proper operation and then have the valve settings checked by the instructor.

3. Introduce feed to the top of the upper tube sheet, and then mark this level with a piece of drafting tape on the gage glass to aid in maintaining this level during the run.

4. Place the evaporator in operation with the calandria under a steam pressure of about 15 pounds per square inch gage and the vapor condensate receiver under vacuum.

5. When conditions have become steady and some salt has crystallized, rapidly remove the salt from the settler, bring the liquor level up to its original mark, determine the quality of the steam, read the barometer, and then start a timed run. Check the quality of the steam again at the end of the timed run.

6. During the timed run, take the following readings about every five to ten minutes for at least an hour:

- a. Time.
- b. Feed to evaporator: gallons or pounds.
- c. Reading of steam condensate receiver: inches on gage glass.
- d. Reading of vapor condensate receiver: inches on gage glass.
- e. Steam pressure: pounds per square inch gage.
- f. Reading of vacuum gage on vapor head: inches of mercury.
- g. Temperature of feed: °F.
- h. Temperature of boiling solution: °F.

7. At the end of the timed run, make sure that the liquor in the evaporator body is at its original mark; and then draw the liquor from the settler into the evaporator, remove the salt, and determine its weight, volume, and percentage of solids. In calculating the quantity of water evaporated, a correction must be applied for the volume of the settler occupied by the salt, since the salt displaces an equivalent volume of solution.

8. Before leaving the laboratory, remove the remainder of the solution from the evaporator, and store it as directed by the instructor. Remove the steam condensate, and pump it into the large tank for pure water storage. Flush out the evaporator and salt settler with water drawn from the vapor condensate tank.

Typical Data from a Student Report. Experimental data for a one-hour timed run during the evaporation of a saturated sodium chloride brine are given in Table 1, and the calculated results are given in Table 2.

Before beginning this test, the heating tubes were examined and found

to be covered with a film of salt; the tubes were also examined at the end of the run and found to be covered with a much thicker layer of salt.

TABLE 1

Experimental Data for the Evaporation of Saturated Sodium Chloride Brine in a Single-Effect Salting-Type Evaporator

Time, P.M.	Steam condensate, in.	Vapor condensate, in.	Steam pressure, lb./in. ²	Evaporator vacuum, in. Hg	Evaporator temperature, °F.
2:00	0	0	15.0	24.7	142
2:10	3.2	1.7	15.0	24.7	142
2:20	6.4	4.5	15.0	25.3	142
2:30	10.4	7.0	15.0	24.8	142
2:40	12.5	8.6	15.0	25.0	142
2:50	15.5	11.0	15.0	25.0	142
3:00	18.7	13.5	15.0	24.9	142
Average.....			15.0	24.9	142

Barometer reading, in. of mercury	= 29.4
Weight of brine fed during run, lb.	= 235.8
Temperature of feed, °F.	= 72
Specific gravity of feed at 60°/60°F.*	= 1.20
Salt in feed, per cent**	= 26.3
Quality of steam, per cent	= 100
Weight of damp salt removed from settler, lb.	= 86.9
Weight of equivalent volume of brine feed, lb.	= 59.7
Solids in damp salt, per cent**	= 72.5
Area of heat transfer surface, ft. ²	= 23.8
Calibration of steam-condensate tank, lb./in.	= 15.8
Calibration of vapor-condensate tank, lb./in.	= 16.6

* Determined by hydrometer.

** Determined by drying.

Sample Calculations.

Item 1*: *Water in brine to evaporator, lb.*

$$\begin{aligned}
 &= \text{total lb. of brine to evaporator} \times \text{fraction of water} \\
 &= \text{Item 3} \times (100 - \% \text{ salt in feed})/100 \\
 &= 296 \times 0.737 = 218
 \end{aligned}$$

Item 2: *Salt in brine to evaporator, lb.*

$$\begin{aligned}
 &= \text{total lb. of brine to evaporator} \times \text{fraction of salt} \\
 &= 296 \times 0.263 = 78
 \end{aligned}$$

* Item numbers refer to quantities in Table 2; all quantities are on an hour basis.

Item 3: *Total brine to evaporator, lb.*

= lb. of brine fed during run + lb. of brine displaced by product in settler

= 236 + 59.7 = 295.7 (or, for practical purposes = 296)

TABLE 2

Calculated Results for the Evaporation of a Saturated Sodium Chloride Brine in a Single-Effect Salting-Type Evaporator

Material Balance (Basis: lb./hr.)			
In		Out	
1. Water	218	4. Vapor condensate	224
2. Salt	78	5. Salt	63
3. Total	296	6. Total	287

Heat Balance			
Datum conditions: 142°F. and liquid state.			
Steam pressure: 15.0 lb./in. ² gage = 29.5 lb./in. ² abs.			
Pressure in evaporator: 4.5 in. Hg abs.			
Boiling temperature of water at 4.5 in. Hg abs.: 130°F.			
Boiling temperature of saturated brine at 4.5 in. Hg abs.: 142°F.			
Heat in, B.t.u./hr.		Heat out, B.t.u./hr.	
7. Steam	312,000	10. Steam condensate	32,000
8. Feed	-13,000	11. Product	0
9. Total	299,000	12. Vapor	228,000
		13. Radiation loss	39,000
		14. Total	299,000
15. Overall heat-transfer coefficient, U , B.t.u./(hr. \times ft. ² \times °F.) = 93.8			
16. Economy, lb. water evaporated per lb. steam = 0.757			
17. Capacity, lb. water evaporated per hr. = 224			

Item 4: *Vapor condensed, lb.*

= 13.5 in. in tank \times 16.6 lb./in. calibration

= 224

Item 5: *Salt in product, lb.*

= lb. damp salt \times fraction of solids

= 86.9 \times 0.725 = 63

Item 7: *Heat in with steam, B.t.u./hr.*

= $W(H_2 - H_1) = 296(1164 - 110) = 312,000$

where

W = lb. steam = lb. condensate = 18.7 in. in tank
 \times 15.8 lb./in. calibration

= 296

H_2 = enthalpy of steam at 250°F. (29.5 lb./in.² abs.),
B.t.u./lb.

$$= 1164$$

H_1 = enthalpy of liquid water at 142°F., B.t.u./lb. = 110

Item 8: *Heat in feed, B.t.u.*

$$= F(t_2 - t_1)c_p = 236 (72 - 142) \times 0.79 = -13,000$$

where

F = weight of cold feed entering from outside, lb. = 236
(The total feed was 296 lb., but 60 lb. were already at the boiling point because this amount came from the settler where it was displaced by the salt that accumulated.)

t_2 = temperature of feed, °F., = 72

t_1 = temperature of boiling liquor, °F. (datum temperature) = 142

c_p = heat capacity of feed, B.t.u./(lb. \times °F.) = 0.79

Item 10: *Heat out with steam condensate, B.t.u.*

$$= W(H_4 - H_1) = 296 (218 - 110) = 32,000$$

where

W = lb. steam = 18.7 in. in tank \times 15.8 lb./in. = 296

H_4 = enthalpy of water at 250°F., B.t.u./lb. = 218
(The condensate was removed at substantially the condensing temperature.)

H_1 = enthalpy of liquid water at 142°F., B.t.u./lb. = 110

Item 11: *Heat out with product, B.t.u.*

= 0 (Product remained substantially at the datum temperature.)

Item 12: *Heat out with vapor, B.t.u.*

$$= \text{Item 4} \times \lambda = 224 \times 1020 = 228,000$$

where

λ = latent heat of evaporation at 130°F., B.t.u./lb. = 1020
(The latent heat was taken at 130°F., the boiling temperature of water, to partially compensate for the increased latent heat of evaporation of water from the solution.)

Item 13: *Radiation, B.t.u.*

$$\begin{aligned} &= \text{Item 7} + \text{Item 8} - (\text{Item 10} + \text{Item 11} + \text{Item 12}) \\ &= 312,000 + (-13,000) - (32,000 + 0 + 228,000) = 39,000 \end{aligned}$$

Item 15: *Overall heat-transfer coefficient, U , B.t.u./($\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}$)*

$$= \frac{Q/\theta}{A(t_3 - t_1)} = \frac{241,000}{23.8 \times (250 - 142)} = 93.8$$

where

$$Q/\theta = \text{heat transferred, B.t.u./hr.} = \text{Item 12} - \text{Item 8} \\ = 228,000 - (-13,000) = 241,000$$

$$A = \text{area of heat-transfer surface, sq. ft.} = 23.8$$

$$t_3 = \text{temperature of condensing steam, } ^\circ\text{F.} = 250$$

$$t_1 = \text{temperature of boiling liquor, } ^\circ\text{F.} = 142$$

Item 16: *Economy, lb. water evaporated per lb. steam*

$$= \frac{\text{Item 4}}{W} = \frac{224}{296} = 0.757$$

where

$$W = \text{lb. steam condensate (as in Item 7)} = 296$$

Item 17: *Capacity, lb. water evaporated per hour*

$$= \text{Item 4} = 224$$

NOMENCLATURE

A = area of heating surface, ft.^2

c_p = heat capacity, B.t.u./($\text{lb.} \times ^\circ\text{F.}$).

F = weight of feed, lb./hr.

H = enthalpy, B.t.u./lb.

Q/θ = heat transferred, B.t.u./hr.

t = temperature, $^\circ\text{F.}$

U = overall heat transfer coefficient, B.t.u./($\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.}$).

W = weight of steam, lb./hr.

λ = latent heat of evaporation, B.t.u./lb.

A SINGLE-EFFECT FORCED-CIRCULATION EVAPORATOR

Designed and Constructed by
The Department of Chemical Engineering
University of Texas*

Description. This forced-circulation evaporator consists of a heat exchanger, an equilibrium chamber, a modified jet condenser, a pump for circulating the liquor, a make-up tank, and drums for collecting the condenser discharge and the steam condensate. All surfaces in contact with liquid are made of copper, brass, or *Everdur*.** Magnesia or rock-wool insulation is used on all parts of the setup where minimum heat loss is desirable.

A drawing of the assembled evaporator is shown in Figure 1, and details of its construction are shown in Figure 2.

Materials and Cost. A list of the materials used in the construction of this evaporator is given in Table 1; these materials cost about \$510. The labor cost for construction and erection was about \$195.

Notes on Construction. *Equilibrium Chamber.* The flanged equilibrium chamber *E* is made by rolling a 6½ x 5-foot plate of 10-gage *Everdur* metal into a cylinder 2 feet in diameter and 5 feet long and then welding on the ½-inch thick rings at the top and bottom.

Heat Exchanger. The heat exchanger *H* is made from a 6½-foot length of A.P.I. line pipe and 15 copper pipes, ½-inch I.P.S. X 6 feet long; the pipes are brazed into a fixed head at the upper end and a floating head at the lower end. Flanges, cut from ½-inch steel plate, are welded to each end of the exchanger.

The 8-inch elbow *D* is made from a length of 8-gage copper tubing by cutting it at the proper angle into three sections, and brazing the pieces together; flanges cut from ½-inch steel plate are brazed to each end.

Condenser. The spray condenser *C* is made from a 25-gallon oil drum and three sets of perforated ½-inch pipes, welded to a ½-inch header through which the condenser water is admitted. The vapor inlet line is brazed into the side of the drum near the bottom.

Operation. Liquor in the evaporator is circulated through heat exchanger *H* and equilibrium chamber *E* by means of centrifugal pump *P*. The vapors released in the equilibrium chamber are drawn into spray condenser *C* and condensed with a measured quantity of cooling water;

* Reported by W. A. Cunningham.

** An alloy of: 98.25 Cu; 1.5 Si; 0.25 Mn.

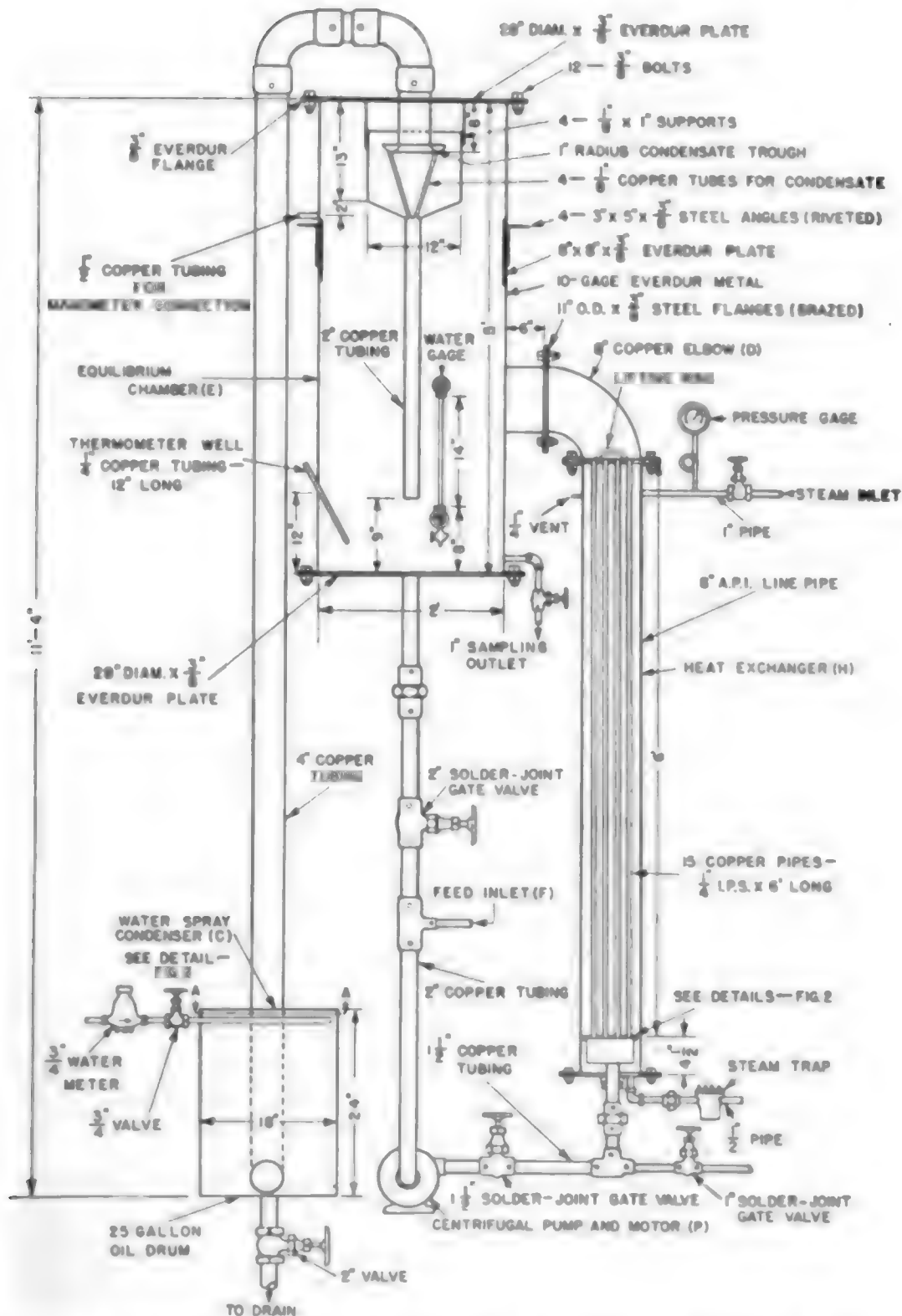


Figure 1. Single-Effect, Forced-Circulation Evaporator—University of Texas

the discharge from the condenser flows into calibrated drums. The increase in the quantity of water collected over the quantity of condenser water used represents the water evaporated. Two drums are used, and a rocker-arm arrangement permits the flow to be alternated between them. Two-inch valves and lines leading to drains permit rapid emptying of these drums.

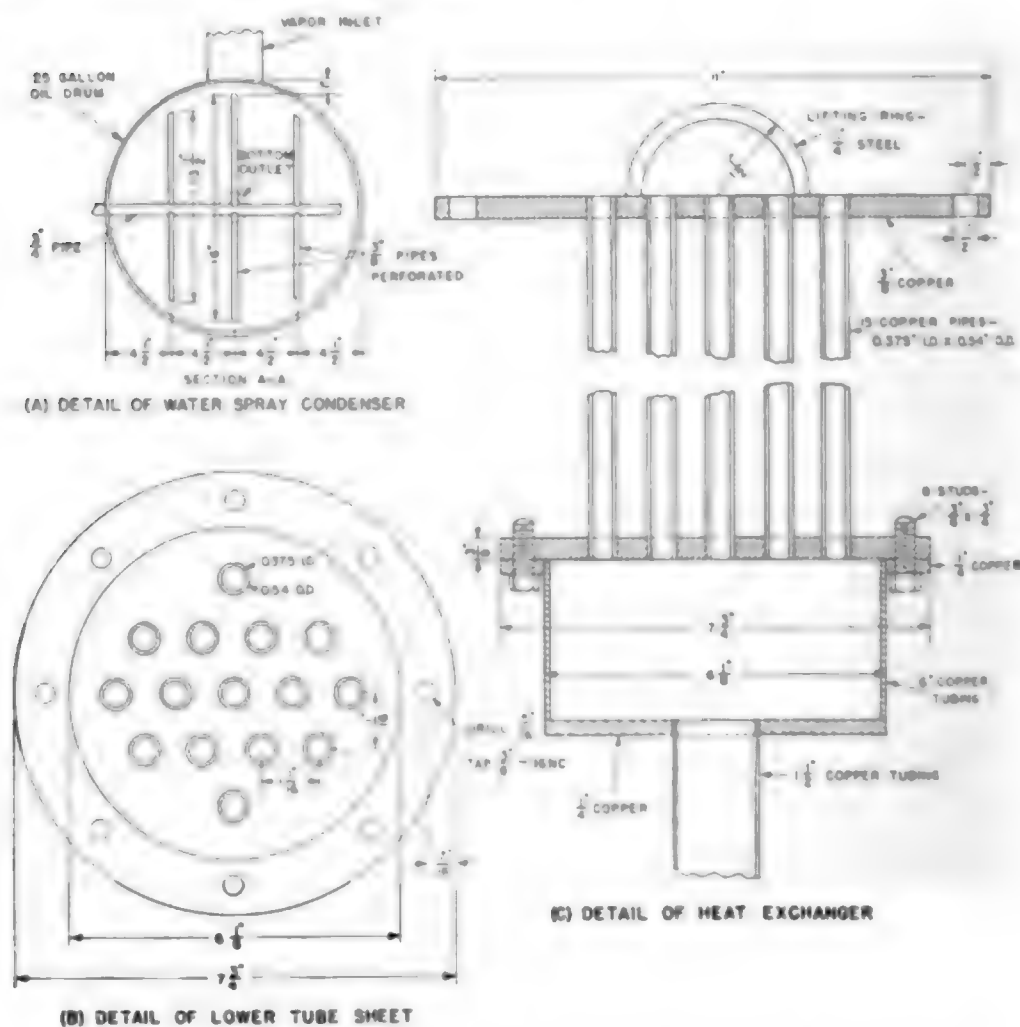


Figure 2. Details of Construction of Forced-Circulation Evaporator

Steam enters the top of the exchanger shell, and the condensate is discharged from the bottom through a steam trap and passes into calibrated drums arranged in the same manner as the drums used to collect the discharge from the spray condenser. The quality of the steam is determined by means of an A.S.M.E. throttling calorimeter.

Feed to the evaporator is first pre-heated in a make-up tank, provided with a steam coil, until its temperature is approximately the same as that in the equilibrium chamber; it is then introduced at F at such a rate that the level in the system remains constant.

TABLE 1

Bill of Materials for Single-Effect Forced-Circulation Evaporator
University of Texas

Quantity	Description	Size	Material
2 ft.	Std. pipe	$\frac{1}{2}$ in.	Black wrought steel
4 ft.	" "	$\frac{1}{2}$ in.	" " "
1	Std. nipple	2 in. x 4 in. long	" " "
1	" "	1 in. x 6 in. "	" " "
1	" "	1 in. x 4 in. "	" " "
2	" "	$\frac{1}{2}$ in. x 4 in. "	" " "
1	" "	$\frac{1}{2}$ in. x 3 in. "	" " "
1	Std. 90° elbow	$\frac{1}{2}$ in.	Black malleable iron
1	Ground-joint union	$\frac{1}{2}$ in.	" " "
1	Plate	6 ft. 4 in. x 5 ft. x 10-gage	Everdur
2	Ring	24 in. I.D. x 28 in. O.D. x $\frac{1}{2}$ in. thick	"
1	Round plate with a 2 $\frac{1}{2}$ -in. hole in center	28 in. O.D. x $\frac{1}{2}$ in. thick	"
1	Round plate with a 4-in. hole in center	28 in. O.D. x $\frac{1}{2}$ in. thick	"
4	Plate	8 x 8 x $\frac{1}{2}$ in.	"
15	Pipe	$\frac{1}{2}$ -in I.P.S. x 6 ft. long	Copper
1	Round plate	7 $\frac{1}{2}$ in. D. x $\frac{1}{2}$ in. thick	"
2	" "	11 in. D. x $\frac{1}{2}$ in. thick	"
1	" "	6 $\frac{1}{2}$ in. D. x $\frac{1}{2}$ in. thick	"
1	" "	13 $\frac{1}{2}$ in. D. x 10 gage	"
1	Ring	4 in. I.D. x 8 in. O.D. x 10 gage	"
2	"	8 $\frac{1}{2}$ in. I.D. x 11 in. O.D. x $\frac{1}{2}$ in. thick	Steel
3	"	8 $\frac{1}{2}$ in. I.D. x 11 in. O.D. x $\frac{1}{2}$ in. thick	"
1	Plate	11 in. O.D. x $\frac{1}{2}$ in. thick	"
8 in.	Rod	$\frac{1}{2}$ in.	"
8 in.	Tubing	12 in.	Copper
3 ft.	"	8 in.	"
3 in.	"	6 in.	"
14 ft.	"	4 in.	"
9 ft.	"	2 in.	"
3 ft.	"	1 $\frac{1}{2}$ in.	"
1 ft.	"	1 in.	"
3 ft.	"	$\frac{1}{2}$ in.	"
1 ft.	"	$\frac{1}{2}$ in.	"

TABLE 1—(Concluded)

Quantity	Description	Size	Material
3 ft.	Flats	1 x $\frac{1}{2}$ in.	Copper
1	Std. gate valve	2 in.	Brass
1	" " "	1 in.	"
1	" " "	$\frac{1}{2}$ in.	"
8	Studs	$\frac{1}{2}$ in.—16NC x $\frac{1}{2}$ in. long	"
32	Bolts and nuts	$\frac{1}{2}$ in.—16NC x $1\frac{1}{2}$ in. long	"
8	" " "	$\frac{1}{2}$ in.—16NC x 2 in. long	"
1	Drum	25-gal. capacity	Steel
1 ft.	Angles	3 x 5 x $\frac{1}{2}$ in.	"
1	Pressure gage	0 to 100 lb., $3\frac{1}{2}$ in. face	Brass
1	Steam-gage siphon	$\frac{1}{2}$ in.	Black wrought steel
1	Steam trap	$\frac{1}{2}$ in.	Cast iron
1	Pump, direct connected to a 1 H.P. splash-proof motor	2-in. suction, $1\frac{1}{2}$ -in. discharge, 30 g.p.m. against a 30-ft. head	All-bronze
1	Water meter	$\frac{1}{2}$ in.	Bronze
1	Water-gage set	$\frac{1}{2}$ in. with 14-in. glass	Brass
8 sq. ft.	Gasket	$\frac{1}{2}$ in. thick	Rubber
6 $\frac{1}{2}$ ft.	Pipe insulation	8 in.	85% magnesia
6 ft.	" "	2 in.	" "
3 ft.	" "	$1\frac{1}{2}$ in.	" "
50 lb.	Insulating cement	—	" "
<i>Solder-joint fittings</i>			
1	Std. tee	2 x 2 x 1 in.	Brass
1	" "	$1\frac{1}{2}$ x 1 x $1\frac{1}{2}$ in.	"
3	Std. 90° elbow	4 in.	"
1	" "	2 in.	"
1	Ground-joint union	2 in.	"
1	" " "	$1\frac{1}{2}$ in.	"
1	Std. gate valve	2 in.	"
1	" " "	$1\frac{1}{2}$ in.	"
2	" " "	1 in.	"

TYPICAL EXPERIMENT

Objects. To determine: (1) the overall heat-transfer coefficient for the evaporation of water in a single-effect, forced-circulation evaporator; and (2) the pounds of water evaporated per pound of steam used.

Auxiliary Equipment. In order to carry out this experiment, the following auxiliary equipment is needed:

1. A.S.M.E. throttling calorimeter.
2. Water meter: $\frac{1}{4}$ -inch.
3. Make-up tank; calibrated.
4. Two drums for collecting the condenser discharge: 55-gallon capacity each; calibrated.
5. Two weighing buckets: 5-gallon capacity each.
6. Scale: 10 kg. capacity.
7. Six thermometers: -10 to 110°C .

Procedure. 1. Fill the evaporator with water to the reference mark on the gage glass of the equilibrium chamber.

2. Prime the pump, turn it on, and circulate the water from the equilibrium chamber through the heat exchanger.

3. Open the steam valve, and adjust the pressure to the desired value.

4. As soon as vapor issues from the condenser outlet, turn on the cooling water to the spray condenser, and adjust the rate of flow of the water so that the vapors are completely condensed without pulling a vacuum of over 1 inch of water on the equilibrium chamber.

5. As evaporation proceeds, add make-up water slowly and continuously through the pump at such a rate as to maintain the liquid level in the gage glass at the reference mark.

6. Allow one hour for the system to attain equilibrium, and then record the following data at five-minute intervals over a period of about 1 to $1\frac{1}{2}$ hours:

- a. The temperatures at the high and low sides of the calorimeter.
- b. The pressure of the steam.
- c. The temperature and rate of flow of cooling water to the condenser.
- d. The temperature and height of water in the drums into which the condenser discharge flows.
- e. The temperature and the height of water in the drums into which the steam condensate flows.
- f. The temperature of the steam condensate.
- g. The temperature of the make-up water.
- h. The height of water in the make-up tank.

7. From these data, calculate the overall heat-transfer coefficient and the pounds of water evaporated per pound of steam used.

Typical Data. The experimental data* for a 55-minute period during

* Experimental data obtained by H. A. Alves.

TABLE 2
Experimental Data for the Determination of the Heat-Transfer Coefficient and the Capacity of Forced-Circulation Evaporator

Barometric pressure: in. of Hg = 29.35
 lb./in.² = 14.42

Calibrations:
 Make-up water tank, gal./in. = 2.205
 Condensate receiver, gal./in. = 1.66

Tube data:
 Inside diameter, in. = 0.375
 Outside diameter, in. = 0.420
 Length, ft. = 6
 Number of tubes = 16

Item	Reading No.											Average value
	1	2	3	4	5	6	7	8	9	10	11	
Steam pressure, lb./in. ² gage.....	14	15	15	15	15	15	15	15	15	15	15	14.9
Calorimeter readings:												
temp. high side, °C.....	127	131	127	130	130	132	135	129	130	131	128	130.9°C. (267.6°F.)
temp. low side, °C.....	121	122	122	124	123	125	127	130	125	126	122	124.2°C. (255.6°F.)
Steam condensed:												
weight, kg./5 min.....	6.05	6.06	6.32	6.36	5.66	5.00	5.00	3.60	3.67	3.64	3.59	5.04 kg. (11.1 lb.)
temp. steam condensate, °C.....	106	106	105	105	105	103	104	101	98	97	96	102.3°C. (216.1°F.)
Temp. water entering exchanger, °C.....	99.5	99.3	99.5	99.5	99.5	99.3	99.5	99.5	99.3	99.5	99.5	99.5°C. (211.1°F.)
Temp. vapors, °C.....	99	99	99	99	99	99	99	99	99	99	99	99°C. (210.2°F.)
Water to condenser:												
gal./5 min.....	46.9	45.0	44.9	45.0	44.8	44.9	44.7	44.0	46.8	55.3	54.2	47.2
temp. °C.....	18.0	18.0	18.0	17.6	17.6	17.5	17.5	17.5	17.5	17.5	17.5	17.6°C. (63.6°F.)
Condenser water plus condensed vapors:												
height in drums, in./5 min.....	29.30	28.60	28.12	28.40	28.12	28.75	28.12	28.60	30.60	20.25	34.00	29.25
temp. °C.....	36.5	36.0	36.0	36.5	37.2	38.0	36.5	39.0	36.5	35.0	34.5	36.9°C. (98.4°F.)
Make-up water added:												
in. in tank.....	—	—	—	2.12	—	—	2.20	—	—	2.12	0.87	0.665 per 5 min.
temp. °C.....	94	—	—	98	—	—	95	—	—	99	98	96.6°C. (205.9°F.)

a run are given in Table 2, and a summary of the calculated results is given in Table 3.

TABLE 3
Calculated Results of Evaporation Experiment

Item No.	Item	Value
1	Make-up water added, lb./hr.....	147.6
2	Condenser water plus condensed vapors, lb./5 min.....	404
3	Condenser water, lb./5 min.....	394
4	Weight of condensed vapors, lb./hr.....	120
5	Total heat given up by steam, B.t.u./hr.....	131,800
6	Heat transferred, based on weight of make-up water added, B.t.u./hr.....	144,090
7	Heat transferred, based on weight of condensed vapors, B.t.u./hr.....	117,080
8	Log-mean area of heating surface, ft. ²	9.34
9	Temperature drop, °F.....	38.8
10	Overall heat-transfer coefficient, based on weight of make- up water added, B.t.u./hr.....	397
11	Overall heat-transfer coefficient, based on weight of vapors condensed, B.t.u./hr.....	323
12	Evaporation, based on weight of make-up water added, lb. water/lb. steam.....	1.11
13	Evaporation, based on weight of condensed vapors, lb. water/lb. steam.....	0.90

Calculations.

Item 1: *Make-up water added, lb./hr.*

$$= 0.665 \times 2.295 \times 231/1728 \times 1/(0.01667) \times 12$$

$$= 147.6$$

where

0.665 = make-up water added, in. in tank per 5 min.

2.295 = tank calibration, gal./in.

231/1728 = ft.³/gal.

0.01667 = specific volume of water at 206.2°F., ft.³/lb.

12 = number of 5-min. periods per hr.

Item 2: *Condenser water plus condensed vapors, lb./5 min.*

$$= 29.35 \times 1.66 \times 231/1728 \times 1/(0.01612)$$

$$= 404$$

where

29.35 = condenser water plus condensed vapors, in. in
tank per 5 min.

1.66 = tank calibration, gal./in.

0.01612 = specific volume of water at temperature in tank, ft.³/lb.

Item 3: *Condenser water used, lb./5 min.*
 $= 47.2 \times 231/1728 \times 1/(0.01604)$
 $= 394$

where

47.2 = cooling water used, gal. per 5 min.

0.01604 = specific volume of water at 63.6°F., ft.³/lb.

Item 4: *Weight of condensed vapors, lb./hr.*
 $= (404 - 394) \times 12 = 120$

where

404 = condenser water plus condensed vapors, lb./5 min.

394 = condenser water, lb./5 min.

Item 5: *Total heat given up by steam, B.t.u./hr.*
 $= 11.1 \times 12 \times (1172.9 - 184.1)$
 $= 131,800$

where

11.1 = steam used, lb./5 min.

1172.9 = enthalpy of steam, B.t.u./lb. (from steam table).

184.1 = enthalpy of steam condensate, B.t.u./lb. (from steam table).

Item 6: *Heat transferred, based on weight of make-up water added' B.t.u./hr.*
 $= 147.6 \times 971.4 + 147.6 (210.2 - 206.2) \times 1.0$
 $= 143,500 + 590 = 144,090$

where

147.6 = make-up water added, lb./hr.

971.4 = latent heat of vaporization of water at 210.2°F., B.t.u./lb.

210.2 = temp. of vapors, °F.

206.2 = temp. of make-up water, °F.

1.0 = heat capacity of water, B.t.u./(lb. \times °F.).

Item 7: *Heat transferred, based on weight of condensed vapors, B.t.u./hr.*
 $= 120 \times 971.4 + 120 (210.2 - 206.2) \times 1.0$
 $116,600 + 480 = 117,080$

where

120 = weight of condensed vapors, lb./hr., and the other values are the same as in Item 6.

Item 8: *Log-mean area of heating surface, ft.²*

$$= \frac{(0.420 - 0.375)}{2.303 \log_{10} \frac{0.420}{0.375}} \times 1/12 \times \pi \times 6 \times 15$$

$$= 9.34$$

where

0.420 = outside diameter of tubes, in.

0.375 = inside diameter of tubes, in.

6 = length of tubes, ft.

15 = number of tubes in evaporator.

Item 9: *Temperature drop, °F.*

$$= 249.0 - 210.2 = 38.8$$

where

249.0 = temperature of saturated steam at 29.32 lb./in.² abs., °F. (from steam table).

210.2 = temperature of vapors, °F.

Item 10: *Overall heat-transfer coefficient, based on weight of make-up water added, B.t.u./(hr. × ft.² × °F.)*

$$= \frac{144,090}{38.8 \times 9.34} = 397$$

where

144,090 = heat transferred, B.t.u./hr.

38.8 = temperature drop, °F.

9.34 = log-mean area of heating surface, ft.²

Item 11: *Overall heat-transfer coefficient, based on weight of vapors condensed, B.t.u./(hr. × ft.² × °F.)*

$$= \frac{117,080}{38.8 \times 9.34} = 323$$

Item 12: *Evaporation, based on weight of make-up water added, lb. water/lb. steam*

$$= \frac{147.6}{11.1 \times 12} = 1.11$$

where

147.6 = weight of make-up water added, lb./hr.

11.1 = weight of steam used, lb./5 min.

Item 13: *Evaporation, based on weight of condensed vapors, lb. water/lb. steam.*

$$= \frac{120}{11.1 \times 12} = 0.90$$

where

120 = weight of condensed vapors, lb./hr., and the other values are the same as in Item 12.

Discussion of Results. There is an appreciable discrepancy between the quantity of make-up water added and the quantity of condensate collected; the former value is apparently too high. However, both of these values may be in error.

The quantity of condensed vapors is determined by the difference between two relatively large numbers—the volume of cooling water plus condensed vapors, and the volume of cooling water alone—and any error in the measurement of these quantities would reflect in a relatively large error in their difference.

The volume of make-up water added to the evaporator may not be a true measure of the total evaporation because it is difficult to bring the final liquor level exactly back to the original value, and a slight difference in level would be equivalent to a comparatively large quantity of liquid.

SECTION 4

DRYING

INTRODUCTION

Although there is no sharp distinction between *drying* and *evaporation*, the term *drying* is usually applied to the removal of relatively small amounts of water from solid materials. There are a number of exceptions to this definition, particularly the drying of liquids in drum and spray driers, but by far the most common type of drying consists in the removal of water from sheet or lump materials by vaporization into air.

Mechanism of Drying. When a slab of wet stock is placed in contact with air, water evaporates from its surface and diffuses as vapor through the air film that covers the surface. The concentration of moisture at the surface is decreased, a concentration difference is set up, and moisture then diffuses from the interior to the surface of the slab,^{6,14,15,16} or it moves to the surface by capillary action.^{6,7} As long as moisture moves to the surface as rapidly as it can vaporize from the surface, the rate of drying is independent of the moisture content of the slab, and is constant, providing that the air velocity, temperature, and humidity do not change. During this constant-rate period, the rate of drying is controlled by the rate of vapor diffusion through the air film and is, therefore, dependent upon the humidity difference across the film and the thickness of the film.

However, when the moisture content becomes less than a certain value, known as the *critical moisture content*, moisture can no longer move to the surface as rapidly as it could be vaporized from the surface, and the drying rate begins to fall and continues to fall with decreasing moisture content until the equilibrium moisture content is reached, and no further drying is possible under the particular drying conditions.

As shown by the typical drying curves of Figure 1, the drying rate during the *falling rate period* may or may not decrease uniformly with decrease in moisture content.* During this period, several factors may be involved, and the relative importance of each is different with different materials and at different stages of drying for the same material.

Immediately below the critical moisture content, there may be a short period of *unsaturated surface drying*; during this period, portions

* The drying rate is generally plotted against free moisture content, which is the total moisture content minus the equilibrium moisture content.

of the surface become dry and, although drying from the still wetted portions continues at the same rate as during the constant rate period, the decrease in wetted surface results in an average decrease in the drying rate.¹¹ The rate during this period is independent of the thickness of the slab, and is frequently proportional to the water content. Furthermore, like the rate during the constant rate period, the rate during unsaturated surface drying is increased by increase in air velocity and decrease in humidity.

As the moisture content of the slab continues to decrease, a point is reached where diffusion through the surface film is no longer of controlling importance, and the drying rate becomes independent of the velocity and the humidity of the air, but is dependent upon the moisture

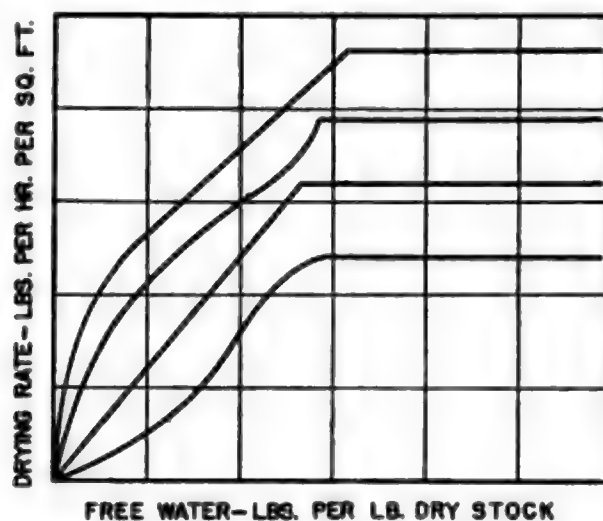


Figure 1. Typical Drying Curves

content and the thickness of the slab. During this period, the drying rate is controlled by the rate at which liquid water diffuses to the surface, or, in the case of granular or fibrous materials where vaporization occurs beneath the surface of the solid, by the rate of vapor diffusion through the zone of vaporization, which moves farther and farther into the interior of the slab as drying progresses.⁹

Heat Transfer During Drying. During drying, the latent heat for vaporization must be transferred to the slab as sensible or radiant heat and, therefore, the slab must be at a lower temperature than the surroundings. If the slab is originally at the same temperature as the surroundings, or at a higher temperature, its temperature will drop as drying begins and will reach an equilibrium value that depends upon the drying conditions.

Drying During the Constant-Rate Period. If heat is supplied only as sensible heat from the drying air and only through the same surfaces from which the slab is drying, the temperature of the slab during the constant rate period will remain at the wet-bulb temperature of the air, for drying takes place, essentially, from a water surface. If the slab is in contact with a heated surface, if heat is flowing from the air through surfaces of the slab where drying is not taking place, or if heat is being radiated to the slab from warmer surroundings, the temperature of the drying surface will be higher than the wet-bulb temperature.

During the constant-rate period, the rate of drying is equal to the rate at which water vapor diffuses through the air film, and this may be expressed by the equation¹

$$\frac{dw}{d\theta} = k_g A (p_w - p_s) \quad (4.1)$$

where

$\frac{dw}{d\theta}$ = rate of diffusion, lb./hr.

k_g = film coefficient of diffusion, lb./(hr. \times ft.² \times atm. partial pressure difference).

A = area of the drying surface, ft.²

p_s = partial pressure of the water vapor in the air, atm.

p_w = partial pressure of the saturated air at the water-air interface, atm. = vapor pressure of water at the surface temperature.

However, since

$$H = \frac{18p}{29(1 - p)} \quad (4.2)$$

where

H = humidity, lb. water/lb. dry air.

p = partial pressure, atm.

18 and 29 = molecular weight of water and air, respectively.

and, under ordinary conditions, p is small compared to 1, Equation (4.2) may be written with only slight error as

$$H = \frac{18}{29} p \quad (4.3)$$

and Equation (4.1) may be written as

$$\frac{dw}{d\theta} = k_g A (H_w - H_s) \quad (4.4)$$

where

$k_{a'}$ = film coefficient of diffusion lb./ (hr. \times ft.² \times atm. humidity difference)

$$= \frac{29}{18} k_a.$$

H_g = humidity of the air, lb. water/lb. dry air.

H_w = humidity of the air at the water-air interface, lb. water/lb. dry air.

The quantities H_w and H_g can be read from a humidity chart, such as Figure 2, or if the stock can be assumed to be at the wet-bulb temperature, the quantity $(H_w - H_g)$ can be calculated from the equation

$$H_w - H_g = \frac{0.26}{\lambda_w} (t_g - t_w) \quad (4.5)$$

where

λ_w = latent heat of vaporization of water at the wet bulb temperature t_w , B.t.u./lb.

$t_g - t_w$ = difference between the dry bulb and wet bulb thermometer readings, °F.

The rate of drying can also be expressed in terms of the heat-transfer coefficient instead of the mass-transfer coefficient since the heat that flows into the slab is equal to the latent heat of vaporization of the water from the slab.

The heat balance for the system is

$$\begin{aligned} \lambda_w \frac{dw}{d\theta} &= \lambda_w k_a A (p_w - p_g) = \lambda_w k_{a'} A (H_w - H_g) \\ &= h_s A_s (t_g - t_s) \end{aligned} \quad (4.6)$$

where

λ_w = latent heat of vaporization of the water, B.t.u./lb.

A_s = surface area through which heat is flowing, ft.²

h_s = mean film coefficient of heat transfer based on A_s , B.t.u./ (hr. \times ft.² \times °F.).

t_g = temperature of the air, °F.

t_s = mean temperature of the surface through which heat is being transferred, °F.

For the common case of adiabatic drying where heat is flowing only through the same area from which drying is taking place,

$$A_s = A \quad (4.7)$$

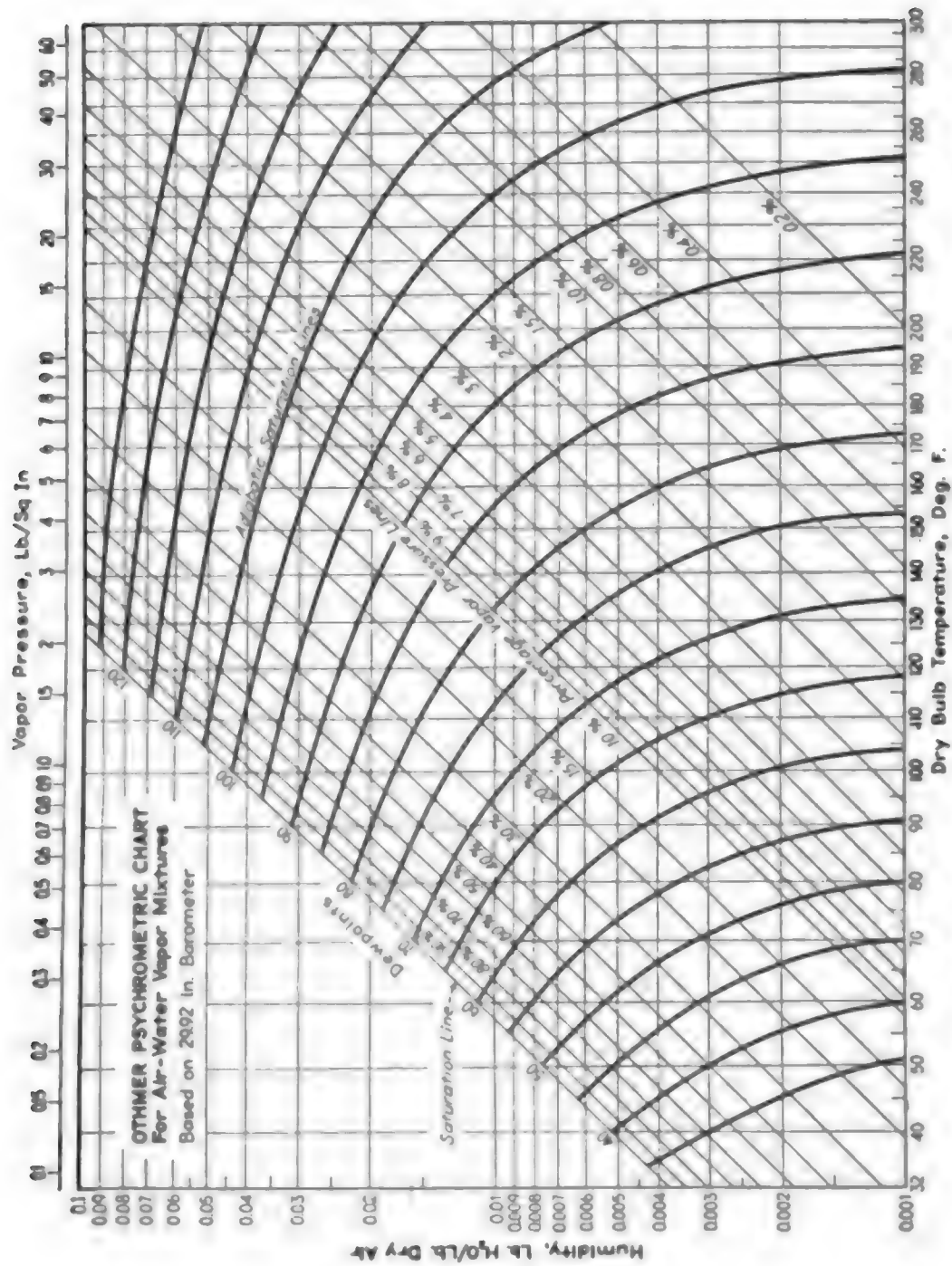


Figure 2. Psychrometric Chart for Air-Water Mixtures (Courtesy of Chem. & Met.)

$$t_s = t_w \quad (4.8)$$

and

$$h_s = h_c \quad (4.9)$$

where

t_w = the wet-bulb temperature of the air, °F.

h_c = coefficient of heat transfer by convection and conduction through the film covering the drying surface, B.t.u./(hr. \times ft.² \times °F.).

Therefore, Equation (4.6) may be written as

$$\frac{dw}{d\theta} = \frac{h_c}{\lambda_w} A(t_g - t_w) \quad (4.10)$$

The magnitude of the film coefficients k_g , k_g' , and h_c depends principally upon the thickness of the air film, and this, in turn, depends largely upon the mass velocity and the direction of flow of the air across the surface. Shepherd, Hadlock and Brewer¹³ have correlated the drying results of several investigators and have found that the data could be expressed reasonably well by the equations

$$k_g = 0.0512 G^{0.76} \quad (4.11)$$

$$h_c = 0.0170 G^{0.76} \quad (4.12)$$

where

G = mass velocity of the air parallel to the surface, lb./(hr. \times ft.²).

Either of these equations, together with Equations (4.1) or (4.10), can be used to estimate the drying rate during the constant rate period providing that the drying takes place at the wet-bulb temperature.

If heat is transferred through surfaces other than the drying surface, or when heat is transferred by radiation, the rate of drying will be increased because of the increased surface temperature. This temperature, which will be different for each specific case, can be estimated by the method outlined by Shepherd, Hadlock, and Brewer.¹³

Drying During the Falling-Rate Period. As moisture may move within a solid by liquid diffusion, vapor diffusion, capillarity, and gravity, it is difficult to estimate the drying time during the falling-rate period except from actual experimental data.

If it is assumed that evaporation takes place only at the surface, that the movement of moisture to the surface is entirely by liquid diffusion, and that the moisture-distribution curve is parabolic, the drying time

during the falling-rate period can be calculated from the general diffusion equation

$$\frac{dw}{d\theta} = -D'A \frac{dc}{dx} \quad (4.13)$$

where

$\frac{dw}{d\theta}$ = rate of movement of water by diffusion.

D' = diffusivity of water.

A = area.

c = concentration of moisture subject to diffusion, in weight per unit volume.

x = distance from midplane in direction of diffusion, where drying is taking place from two surfaces.

On the basis of the above assumptions, integrated drying equations have been developed for cases where resistance to liquid-diffusion is controlling, where surface evaporation is controlling, and where both resistances are important.^{8, 10, 12, 17}

When the experimental drying curve is available, the time required to dry a slab from one moisture content to another during the falling rate period can be calculated by graphically solving the equation²

$$\int_0^{\theta_f} = \frac{D}{A} \int_{F_1}^{F_2} \frac{dF}{f(F)} = \theta_f \quad (4.14)$$

where

θ_f = time of drying, hr.

D = weight of dry solid, lb.

A = drying area, ft.²

F_1 = free moisture content at the beginning of the drying period, lb. water/lb. dry solid.

F_2 = free moisture content at the end of the drying period, lb. water/lb. dry solid.

$f(F)$ = ordinate of the drying-rate curve (Figure 1), where the free moisture content is F lb./lb. dry solid.

In practice, values of the drying rate $f(F)$ are read from the drying-rate curve, and their reciprocals are plotted against the corresponding values of F . The curve of $f(F)$ vs. F is integrated graphically between the desired limits, and the value of the integral is multiplied by D/A to obtain θ_f .

In many cases, it is sufficiently accurate to assume that the drying curve during the falling-rate period is a straight line from the critical

point to the origin. In this case, the drying time can be calculated without resorting to graphical methods, for Equation (4.14) becomes

$$\theta_f = 2.303 \frac{DF_c}{AR_c} \log_{10} \frac{F_1}{F_2} \quad (4.15)$$

where

F_c = free moisture content at the critical point, lb. free water/lb. dry stock.

R_c = rate of drying during the constant rate period, lb./ (hr. \times ft.²).

If the drying rate curve is not a straight line through the origin but is straight from the critical point down to the desired final moisture content, the drying time can be calculated from the equation⁴

$$\theta = \frac{D}{A} \times \frac{(F_c - F_2)}{R_m} \quad (4.16)$$

where

θ = time in hours required to dry from a critical moisture content of F_c down to a final moisture content of F_2 .

$$R_m = \text{log-mean drying rate, lb./ (hr. } \times \text{ ft.}^2) = \frac{R_c - R_2}{2.303 \log_{10} \frac{R_c}{R_2}}$$

and the other symbols have the same units as in previous equations.

Drying Under Variable Conditions. When the drying conditions are not constant, that is, when the air changes in temperature or humidity while passing over the stock, the above equations cannot be used.

If drying is entirely above the critical moisture content, the mean drying rate $\left(\frac{dw}{d\theta}\right)_m$ can be calculated from the equations

$$\left(\frac{dw}{d\theta}\right)_m = k_g A (p_w - p_g)_m = k_g A (\Delta p)_m \quad (4.17)$$

$$= k_{g'} A (H_w - H_g)_m = k_{g'} A (\Delta H)_m \quad (4.18)$$

where

$(p_w - p_g)_m$ and $(H_w - H_g)_m$ are the mean partial pressure difference and the mean humidity difference, respectively, between the conditions at the air-water interface and in the bulk of the air.

If measurements of the temperature and humidity are available throughout the length of the drier, values of $(\Delta p)_m$ or $(\Delta H)_m$ can be plotted against the length of the drier, and the mean value can be determined graphically. If temperatures and humidities at only the terminal conditions are known, the logarithmic-mean value of Δp or ΔH may be used.⁴

During the falling-rate period and where the drying curve can be considered to be a straight line from the critical point through the origin, the drying area A required to reduce the free moisture content from a critical value of F_c down to a final value of F_2 with a change in the humidity of the air from H_c to H_2 can be calculated from the equation

$$A = \frac{F_c}{k_G \left(\frac{H_\infty - H_c}{D} + \frac{F_c}{G} \right)} \times 2.303 \log_{10} \frac{F_c(H_\infty - H_2)}{F_2(H_\infty - H_c)} \quad (4.19)$$

Types of Driers. Driers may be classified into several types:

1. Compartment
2. Tunnel
3. Rotary
4. Agitator
5. Cylinder
6. Drum
7. Spray

Atmospheric compartment driers are enclosures supplied with heated air, which is usually forced over the material being dried by fans or blowers; if the drier consists of more than one compartment, the air is most commonly passed through the compartments in series, being reheated as it leaves one compartment and before it enters the next. If the stock is granular or lumpy, it is dried on trays, while sheet materials are supported on racks so that drying may take place from both sides.

Vacuum tray driers are, ordinarily, single-compartment driers in which heat is supplied, usually as steam, through hollow shelves on which trays rest; the evaporated moisture is removed by means of a vacuum pump and a condenser.

Drying in a compartment drier is a batch operation and, therefore, economical only for the drying of relatively small quantities of materials; when large quantities are to be dried, a *tunnel drier* is preferable. In a tunnel drier, heated air or other hot gas is passed through a tunnel, usually countercurrent to the stock which is carried through continuously on cars or belts.

Rotary driers are rotating, inclined cylinders heated indirectly through the metal walls, directly by means of hot gases which come in contact with the continuously moving stock, or by a combination of both direct and indirect heating. These driers are particularly adapted to the drying of free-flowing granular, crystalline, or lumpy materials.

Agitator driers may be shallow, circular pans with a scraper agitator mounted on a vertical shaft, they may be stationary, horizontal cylinders with agitator blades mounted on a rotating horizontal shaft, or they may be screw conveyors in jacketed troughs; they may be operated at atmospheric pressure or under vacuum. The pan type is a batch drier; the others are usually continuous. In all types, heat is transferred through a metal wall. Agitator driers can handle a wide variety of granular or lumpy materials, but they are particularly adapted for the drying of materials which are too sticky to be handled in rotary driers.

Cylinder driers are used for the continuous drying of paper and textiles; they consist of series of steam-heated rolls over which the stock is carried. *Drum driers* are rotary, steam-heated cylinders, operating either singly or in pairs, which pick up a film of solution, and as they rotate drive off the water and leave a film of dried solid, which is scraped off by a doctor knife. These driers may be either of the atmospheric or vacuum type; the latter type is used principally where it is desired to lower the temperature of drying so as to minimize decomposition of the material being dried.

Spray drying, as the name implies, is accomplished by spraying a solution into heated air or other hot gas. Drying is very rapid because of the large amount of surface exposed and, therefore, spray drying is desirable for many food products or other easily oxidized organic materials. Heat economy, however, is very poor because of the high sensible-heat loss in the waste gas; therefore, spray drying is only practical where the value of the product is relatively high.

Laboratory Driers. Although laboratory driers may be of any type, the most practical is a compartment drier with provision for controlling the temperature, the humidity, and the velocity of the air over the stock. The number of compartments is of little importance; one compartment is usually sufficient and makes it easier to weigh the stock without disturbing the drying conditions, for the stock can be suspended from the beam of a balance mounted on top of the drier.

Errors in Laboratory Experiments. The object of laboratory drying tests is to determine drying rates or coefficients for use in the design of

large-scale equipment, or as an aid in predicting the performance of large-scale equipment. For either purpose, the results must be reasonably accurate, but, unfortunately, many laboratory drying tests are in error because the temperature of the drying surface is not known.

It is common practice to assume that the stock is at the wet-bulb temperature of the air and to calculate the driving force in terms of partial pressure or humidity difference on the basis of the wet- and dry-bulb thermometer readings. This is justified only when the heat for vaporization comes only as sensible heat from the drying air and is transferred only through the same surface from which drying is taking place.

If the material is dried in uninsulated pans—as is frequently the case—considerable heat may be transferred through the pan itself; the stock may also receive appreciable heat by radiation from other trays or pans, or even from the heating coils. In such cases, the drying coefficients determined on the basis of the wet-bulb temperature will be low, perhaps by a considerable amount. To minimize this error, pans should be well insulated and placed in the drier so as not to receive radiant heat. It is also advisable, although seldom done, to attach thermocouples to the surface of the stock and to base the calculations of the coefficient on a true, instead of on an assumed, temperature.

NOMENCLATURE

- A = area, ft.²
- c = concentration, weight/unit volume.
- D = weight of dry solids, lb.
- D' = diffusivity.
- F = free moisture content, lb. water/lb. dry solid.
- $f(F)$ = ordinate of drying-rate curve, lb./lb. dry solids.
- G = mass velocity, lb./(hr. \times ft.²)
- h or h_c = film coefficient of heat transfer, B.t.u./(hr. \times ft.² \times °F.).
- H = humidity, lb. water/lb. dry air.
- ΔH = humidity difference, lb. water/lb. dry air.
- k_σ = film coefficient of diffusion, lb./(hr. \times ft.² \times atm. partial pressure difference).
- $k_{\sigma'}$ = film coefficient of diffusion, lb./(hr. \times ft.² \times atm. humidity difference).
- p = partial pressure, atm.
- Δp = partial pressure difference, atm.
- R = rate of drying, lb./(hr. \times ft.²)
- t = temperature, °F.
- $\frac{dw}{d\theta}$ = rate of diffusion, lb./hr.

x = distance through which diffusion takes place.

λ = latent heat of vaporization, B.t.u./lb.

θ = time of drying, hr.

Subscripts

1 = beginning of drying period.

2 = end of drying period.

c = critical point, except in h_c .

f = final.

g = air.

m = mean.

s = surface through which heat is flowing.

w = water-air interface.

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A COMPARTMENT TRAY DRIER

Designed and Constructed by

The Department of Chemical Engineering
University of North Dakota

Description. This is a steam-heated, five-pass compartment drier, 6 feet wide, 2 feet deep, and 5 feet high, and with a total drying surface of approximately 33 square feet. The framework is made of welded steel angles and channels; the sides, top, bottom, and trays are made of galvanized sheet steel; the external air ducts are galvanized stove pipe; the heating units—one at each end—which provide a total heating surface of about 13.9 square feet, are made by welding a number of $\frac{1}{2}$ -inch standard black steel pipes to 2-inch headers. All external walls are insulated with 1 inch of mineral wool.

Air is circulated over the trays and heating units by a conoidal ventilating fan, and provision is made for measuring its velocity with an anemometer, and its temperature and humidity, as it enters and leaves each compartment, with dry- and wet-bulb thermometers.

A view of the completed drier is shown in Figure 1.

Materials and Cost. A list of the materials used in the construction of this drier is given in Table 1. These materials cost approximately \$125.

Auxiliary Equipment. In carrying out an experiment for the determination of drying rates, the following additional equipment is needed:

1. Anemometer or pitot tube for measuring the air velocity.
2. Stop watch.
3. Balance: a single- or double-beam platform balance having a capacity of 1000 grams and reading to 0.1 gram is satisfactory.
4. Wet- and dry-bulb thermometer set, as shown in Figure 2.
5. Thermometers, or copper-constantan thermocouples and a potentiometer: Fahrenheit thermometers with a range of 20 to 220° and graduated to 1° are adequate.

Notes on Construction. This drier is most easily constructed in the following manner, as illustrated in Figures 3 and 4:

1. Cut the angles for the inner framework and the supports for the shelves, drill and tap them to take the $\frac{3}{8}$ -inch stove bolts used to attach the sheet steel, and weld them together as shown in Detail A.
2. Cut the 16-gage sheet steel for the four upper shelves, drill the

necessary bolt holes and the holes for the heating pipes, and bolt the shelves in place.

3. Insert the $\frac{1}{2}$ -inch heating pipes through the holes provided, and braze them to the shelves.

4. Drill the necessary holes in the headers, weld on the $\frac{1}{2}$ -inch nipples for the air vents and the $\frac{1}{2}$ -inch nipples for the condensate lines, and weld the headers to the heating pipes. Detail *B* shows the unit at this stage of construction.

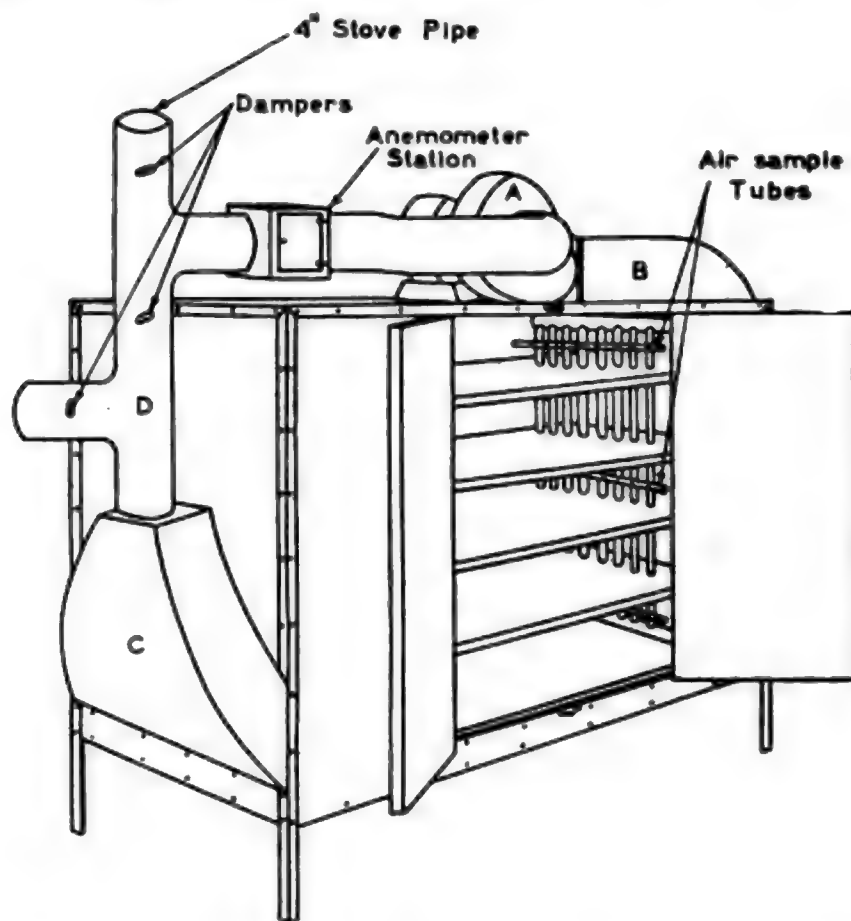


Figure 1. Compartment Tray Drier—University of North Dakota

5. Cut and drill the 16-gage sheet steel for the bottom of the inner section and the 24-gage steel for the top, sides, and front, providing openings for the doors, air ducts, and air-sample tubes; bolt these sheets in place, and braze on the nipples for the air-sample tubes. To insure tightness, insert felt strips along the edge of each shelf before bolting on the sides. This stage of the construction is shown in Detail *C*.

6. Make and attach the air ducts as shown in Detail *D*.

7. Construct the floor of the outer supporting structure shown in

TABLE 1
Bill of Materials for Compartment Tray Drier
 University of North Dakota

Quantity	Description	Size	Material
10 ft.	Std. pipe	2 in.	Black wrought steel
50 ft.	" "	$\frac{1}{2}$ in.	" " "
2	Std. nipple	$\frac{1}{2}$ in. x 5 in. long	" " "
20	" "	$\frac{1}{2}$ in. x 2 in. "	" " "
2	" "	$\frac{1}{2}$ in. x 6 in. "	" " "
1	Std. tee	2 in.	Black malleable iron
1	" "	$\frac{1}{2}$ in.	" " "
1	Std. 90° elbow	2 in.	" " "
1	" " "	$\frac{1}{2}$ in.	" " "
1	Ground-joint union	2 in.	" " "
1	" " "	$\frac{1}{2}$ in.	" " "
1	Std. gate valve	2 in.	Brass
2	Std. globe valve	$\frac{1}{2}$ in.	"
82 ft.	Angles	1 x 1 x $\frac{1}{2}$ in.	Steel
95 ft.	"	$\frac{1}{2}$ x $\frac{1}{2}$ x $\frac{1}{2}$ in.	"
17 ft.	Channels	2 x 1 x $\frac{1}{8}$ in.	"
19 ft.	"	1 x $\frac{1}{2}$ x $\frac{1}{2}$ in.	"
5.5 ft.	Flats	1 x $\frac{1}{2}$ in.	Cold rolled steel
70 sq. ft.	Sheet	2 ft. wide x 16-gage	Galv. steel
60 sq. ft.*	"	3 ft. wide x 24-gage	" "
78 sq. ft.	"	20 x 28 in. x 30-gage	" "
2 lengths	Stove pipe	4 in. diam. x 24 in. long	" "
2	Stove-pipe 90° elbow	4 in.	" "
3	Stove-pipe damper	4 in.	Cast iron
1	Steam trap	$\frac{1}{2}$ in.	" "
1	Pressure gage	0 to 60 lb. press., 3 $\frac{1}{2}$ -in. face	Brass
1	Steam-gage siphon	$\frac{1}{2}$ in.	Black wrought steel
2	Sash fastener	$\frac{1}{4}$ x 2 $\frac{1}{2}$ in.	Brass
2	Sash lift	1 $\frac{1}{2}$ x 4 in.	"
2 pair	Butt hinge	2 $\frac{1}{2}$ x 1 $\frac{1}{2}$ in.	Steel
300	Stove bolt	$\frac{1}{8}$ x $\frac{1}{2}$ in.	"
25 ft.	Tubing	$\frac{1}{2}$ in.	Copper
1	Conoidal fan and motor	98 cu. ft. free air per min.	—
30 ft.	Felt strip	1 x $\frac{1}{2}$ in.	Cotton
5 cu. ft.	Insulation	—	Rock wool

* This includes sufficient material for the construction of 5 pans each 1 ft. x 1 ft. x 1 in. deep.

Detail E, block it up 10 $\frac{1}{2}$ inches from the floor, cover it with mineral wool insulation, and set the constructed inner section in place.

8. Construct the outer framework around the inner section, and cover the sides, but not the top, with sheet metal. The construction of the outer framework is shown in Detail *F*.

9. Insert the nipples for the air-sample tubes into the openings at the back of the drier and braze them to the outer sheet. The location and arrangement of the sample tubes are shown in Details *G* and *H*.

10. Fill the annular space with insulation.

11. Fit the outer top in place.

12. Construct and mount the two doors as shown in Detail *I*.

13. Weld the blower supports in place, mount the blower, and assemble the external air duct, including the anemometer station.*

14. Cover the corners of the outer framework with steel angles.

15. Drill a series of $\frac{1}{8}$ -inch holes along one side of each of the $\frac{1}{2}$ -inch copper air-sampling tubes, insert the tubes through the nipples, and make the openings between the tubes and the nipples air tight with asbestos rope or other suitable material.

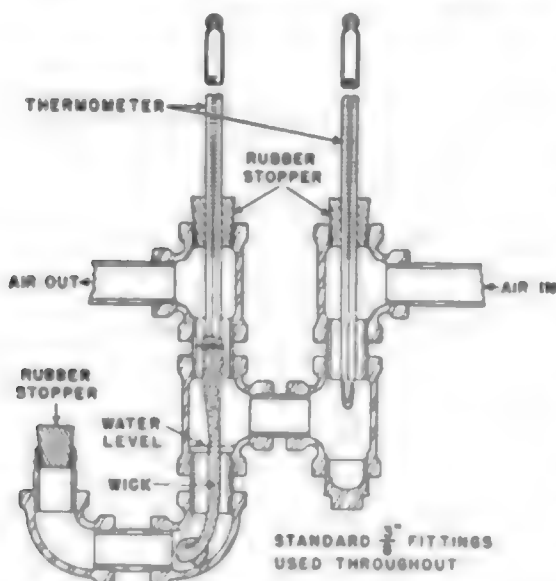


Figure 2. Wet- and Dry-Bulb Thermometer Set

Comments on Design. *Continuous Weighing.* Removing the samples from the drier to make periodic weighings upsets the drying conditions and is unsatisfactory where it is desired to obtain accurate data. The drier can be modified very easily to permit direct continuous

weighings when only one pan of material is to be dried. This is done by mounting a balance on the top of the drier, attaching a wire to the bottom of the left pan, and passing it through small openings drilled through the top and the shelves of the drier at approximately their centers. A pan containing the stock, or the stock itself if solid slabs are used, is attached to the lower end of the wire.

Humidity Control. During a drying experiment, the humidity of the laboratory may vary appreciably and thus affect the accuracy of the results. Furthermore, it may be desirable to carry out drying experiments under conditions of relatively high humidity.

* The anemometer station can be eliminated if it is preferred to measure the air velocity by means of a pitot tube.

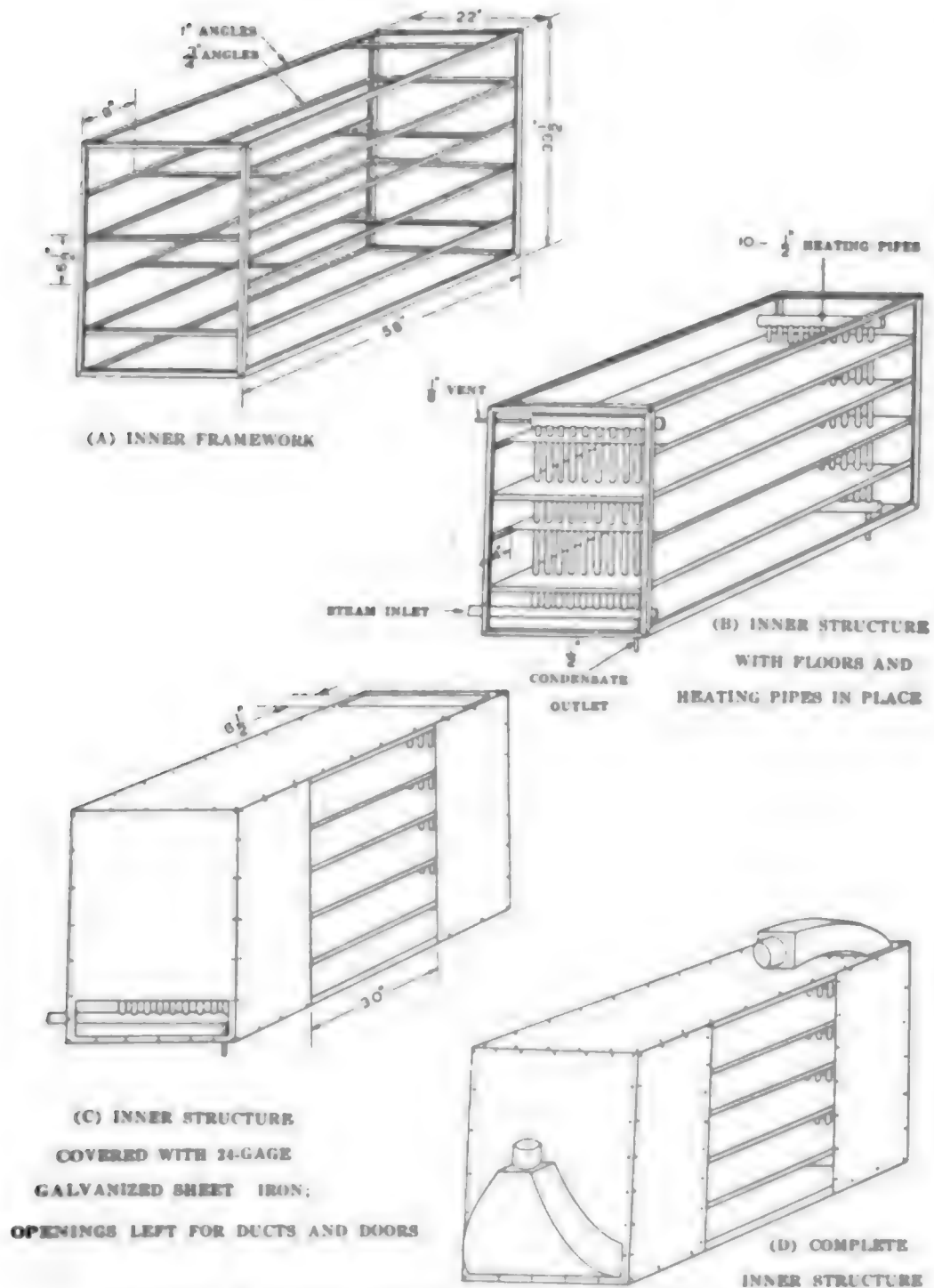


Figure 3. Stages in the Construction of Compartment Tray Drier

An automatically-controlled humidification unit for use with the tray drier is shown in Figure 5. This humidifier, which has given excellent results, was designed by the Department of Chemical Engineering of the *University of New Hampshire* and constructed at a cost of about \$50. A bill of materials for this humidifier is given in Table 2.

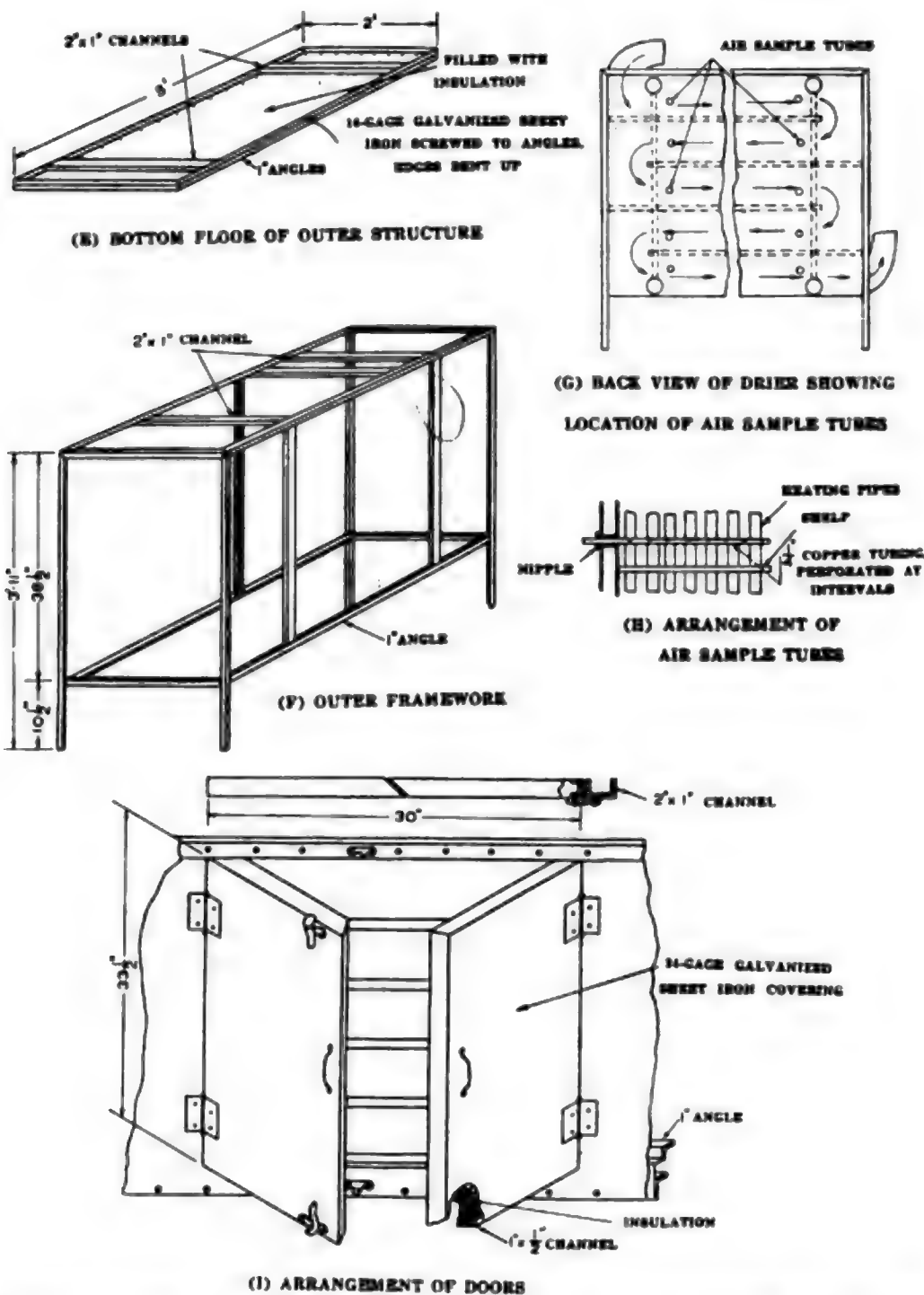


Figure 4. Stages in the Construction and Details of Compartment Tray Drier

Humidification unit is operated as follows:

1. The humidistat *A* is set for the desired humidity.
2. Water is introduced continuously through spray nozzle *B* at a rate just insufficient to maintain the desired humidity.

3. Air valve *C* and water valve *D* are opened, and electrical switch *E* is closed.

When the humidity falls below the value for which the humidistat is set, the magnetically-operated valve *F* opens, and an air-water mixture is injected into chamber *G*.

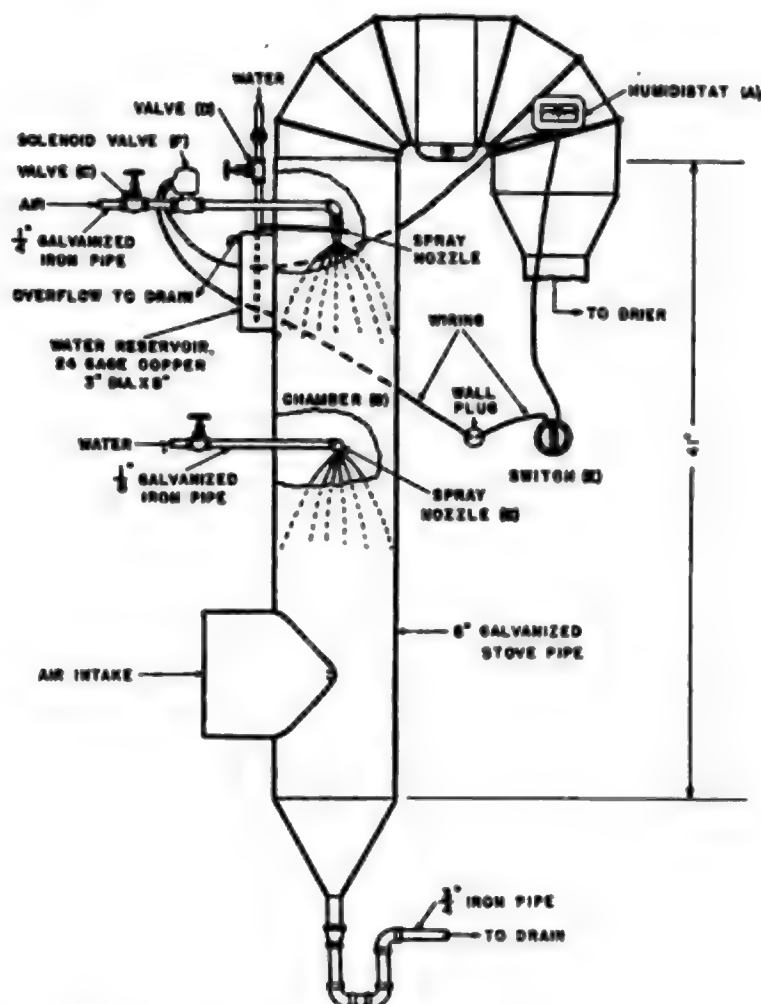


Figure 5. Humidification Unit for Compartment Tray Drier—University of New Hampshire

Operation of the Drier. Air, from blower *A* (Figure 1), enters the drier through duct *B*, circulates successively through each of the five compartments, and leaves through duct *C*. As the air passes from one compartment to the next, it is reheated by the steam coils located at each end of the drier.

The air leaving the drier may be discarded, or any part of it may be recirculated by adjusting the dampers in air duct *D*.

The material is dried in pans that are placed on the drier shelves. For experimental determination of drying rates, the pans are removed and weighed at intervals, and wet- and dry-bulb temperatures are measured periodically.

TYPICAL EXPERIMENT

Object. To determine the rate of drying of wet sawdust under different conditions of temperature and air velocity.

TABLE 2
Bill of Materials for Humidifier
University of New Hampshire

Quantity	Description	Size	Material
4 ft.	Stove pipe	8-in. x 26-gage	Galv. steel
2	Stove-pipe elbow	8 in. x 26-gage	" "
1	Std. nipple	$\frac{1}{2}$ x 3 in.	Galv. wrought steel
2	Std. 90° elbow	$\frac{1}{2}$ in.	Galv. malleable iron
2	Std. globe valve	$\frac{1}{2}$ in.	Brass
1	" " "	$\frac{1}{2}$ in.	"
1	Spray nozzle*	No. $\frac{1}{2}$ A-5; 0.5 gal. water per min. at 10 lb. air press.	"
1	Pneumatic atomizing nozzle*	No. 1; 0.27 gal. water per hr. at 10 lb. air press.	"
1	Solenoid valve**	No. 68 SR3; $\frac{1}{4}$ in. orifice, 110 volts, 60 cycle, 15 watts	—
1	Insertion type humidistat***	—	—
1	Electric switch	110 volts	—

* Mfgd. by Spraying Systems Co., Chicago, Ill.

** Mfgd. by Detroit Lubricator Co., Detroit, Mich.

*** Mfgd. by Friez Co., Baltimore, Md.

Procedure. 1. Measure:

- a. The free cross-sectional area of each compartment with the pans in place.
 - b. The surface area of each pan.
 - c. The cross-sectional area of the duct at the anemometer station.
2. Adjust the temperature and velocity of the air to the desired values by regulating the steam pressure and setting the dampers to proper openings.
 3. After conditions have become constant, place a weighed pan of wet sawdust, containing approximately 75 per cent water, on each shelf.

4. Make the following measurements at frequent intervals:
 - a. Velocity of the air.
 - b. Wet- and dry-bulb temperatures of the air entering and leaving the drier and before and after it passes over each pan.
 - c. Weight of each pan of material.*
5. Continue drying until there is no further loss in weight.
6. Repeat the experiment under different conditions of air velocity and steam pressure.
7. For each run, calculate the drying rates for the material on each shelf, and plot these rates as a function of the free moisture content.
8. For each sample, calculate the drying coefficient k_d for the constant rate period by means of the equation¹

$$\frac{dw}{d\theta} = k_d(H_w - H_a)A \quad (1)$$

Typical Data. The following data and calculations were taken from a student report. Only the results obtained from one pan of material during one run are presented here, although the complete report included data and calculations for five pans during two runs.

The experimental data for Run No. 1 are given in Table 3; these data were analyzed in the following manner:

1. The weight of the sample was plotted against time, as shown in Figure 6. This curve was then graphically differentiated to obtain the drying rate curve shown in Figure 7; the calculations involved in this differentiation are given in Columns 1 to 5 of Table 4.

2. The values of the drying rates $\frac{dw}{d\theta}$ in grams of water evaporated per minute at various times θ were read from Figure 7 and listed in Column 6 of Table 4. These rates were converted into the units of lb./hr. \times ft.² by multiplying them by the factor:

$$\frac{\text{min./hr.}}{\text{grams/lb.}} \times \frac{1}{\text{pan area, ft.}^2} = \frac{60}{454} \times \frac{1}{1} = 0.0321$$

3. The drying rates, which are given in Column 7 of Table 4, were plotted against the free water content, in lb. per lb. of dry sawdust, in Figure 8. The free water (Column 9 of Table 4) was calculated from the weight of the sample at time θ (Column 2 of Table 4) and the weight of dry sawdust (last figure in Column 12 of Table 3). This calculation is:

* It is essential that removal of pans and weighings be performed rapidly to minimize disturbance of the drying conditions.

$$\frac{\text{wt. sample} - \text{wt. dry sawdust}}{\text{wt. dry sawdust}} = \frac{\text{wt. sample in grams} - 94.6}{94.6}$$

TABLE 3

Experimental Data for the Drying of Wet Sawdust in a Compartment Drier
(Run No. 1; Compartment No. 2)

- (1) Weight of empty pan, grams = 750
 (2) Weight of pan plus wet sawdust, grams = 450
 (3) Weight of wet sawdust, grams = 300
 (4) Cross-sectional area of compartment with pan in place, ft.² = 0.826

Time	Elapsed time, min.	Air. temp., °F.				Compartment temp., °F.				Exit air velocity, ft./min.	Weight of sample, grams
		Inlet		Exit		Left Side		Right Side			
		Dry bulb	Wet bulb	Dry bulb	Wet bulb	Dry bulb	Wet bulb	Dry bulb	Wet bulb		
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)
8:42 A.M.	—	81.0	67.0	—	—	131.0	86.0	130.8	87.2	—	—
8:57*	0	—	—	164.0	91.8	—	—	—	—	473	300.0
9:16	19	81.5	67.5	161.0	92.8	134.0	86.0	130.0	86.4	474	—
9:31	34	81.0	67.0	—	—	—	—	—	—	—	268.2
9:56	59	81.5	67.3	—	—	133.5	85.7	129.0	84.5	—	—
10:02	65	—	—	—	—	—	—	—	—	—	237.7
10:25	88	—	—	162.0	92.5	132.5	85.5	129.0	84.6	—	—
10:31	94	81.5	67.5	162.5	92.0	—	—	—	—	—	210.0
10:46	109	—	—	—	—	—	—	—	—	—	196.6
11:00	124	81.0	66.9	162.6	—	132.5	85.0	129.0	85.3	—	181.4
11:16	139	—	—	—	—	—	—	—	—	—	170.4
11:30	153	80.0	65.9	161.4	—	133.0	85.0	130.0	84.0	464	—
11:36	159	—	—	—	—	—	—	—	—	—	156.3
12:00	183	80.0	65.5	163.4	—	133.0	84.6	130.0	84.3	467	—
12:06 P.M.	189	—	—	—	—	—	—	—	—	—	137.8
12:30	213	80.0	65.5	163.0	91.4	133.0	84.7	130.0	84.4	473	—
12:45	228	—	—	—	—	—	—	—	—	—	117.2
1:15	258	—	—	—	—	—	—	—	—	—	107.8
1:40	283	80.5	66.0	164.4	93.4	135.5	85.6	132.2	84.8	472	—
1:45	288	—	—	—	—	—	—	—	—	—	100.0
2:00	303	—	—	—	—	—	—	—	—	—	99.4
2:18	321	—	—	—	—	—	—	—	—	—	97.3
9:00**	723	—	—	—	—	—	—	—	—	—	94.6

* Pan introduced at this time. Each pan was 1 ft. square.

** There was no further loss in weight after this time.

CALCULATIONS

Drying Coefficient During the Constant-Rate Period. This is calculated from Equation (1), which may be written as

$$k_{G'} = \frac{dw}{Ad\theta} \left(\frac{1}{H_w - H_g} \right) \quad (2)$$

where

$\frac{dw}{Ad\theta}$ = drying rate, lb. water evaporated per hr. per ft.²

$k_{G'}$ = drying coefficient, lb. water evaporated per hr. per ft.² per unit humidity difference.

TABLE 4

Summary of Calculations of the Drying Rates and Free Water Contents

Elapsed time θ , min.	Weight of sample, grams	$\Delta\theta$, min.	Δw , grams	$\frac{\Delta w}{\Delta\theta}$, grams/min.	$\frac{dw}{d\theta}$, grams/min.	Drying rate, lb./ (hr. \times ft. ²)	Free water, grams	Free water, lb./lb. dry stock
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
0	300				0.96	0.127	205.4	2.173
124	181	124	119	0.96	0.96	0.127	86.4	0.913
130	175.5	6	5.5	0.91	0.85	0.112	80.9	0.855
140	167.5	10	8.0	0.80	0.77	0.102	72.9	0.771
150	160.0	10	7.5	0.75	0.71	0.094	65.4	0.691
160	153.0	10	7.0	0.70	0.65	0.086	58.4	0.618
180	141.0	20	12.0	0.60	0.56	0.074	46.4	0.491
200	131.0	20	10.0	0.50	0.49	0.065	36.4	0.385
220	121.5	20	9.5	0.475	0.44	0.058	26.9	0.285
240	113.0	20	8.5	0.425	0.37	0.049	18.4	0.195
260	106.0	20	7.0	0.35	0.29	0.038	11.4	0.121
280	101.0	20	5.0	0.25	0.20	0.026	6.4	0.068
300	98.0	20	3.0	0.15	0.11	0.015	3.4	0.036
320	96.5	20	1.5	0.075	0.05	0.007	1.9	0.021

H_w = saturation humidity of air at the wet-bulb temperature, lb. water per lb. dry air.

H_g = humidity of the drying air, lb. water per lb. dry air.

The humidity difference ($H_w - H_g$) is calculated from the equation²

$$(H_w - H_g) = \frac{0.26}{\lambda_w} (t_g - t_w) \quad (3)$$

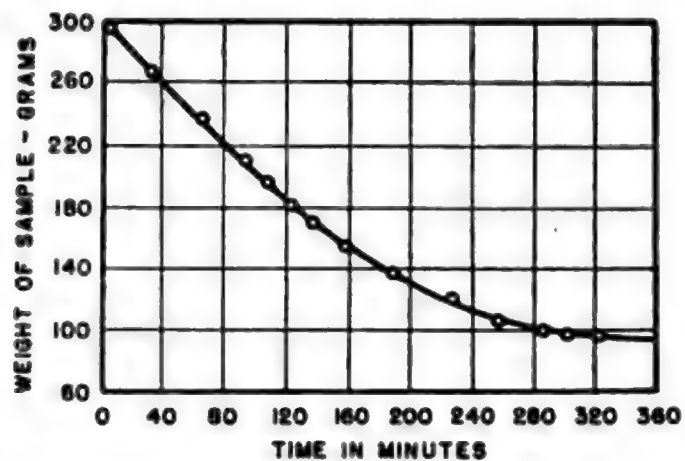


Figure 6. Weight of Sawdust as a Function of Drying Time

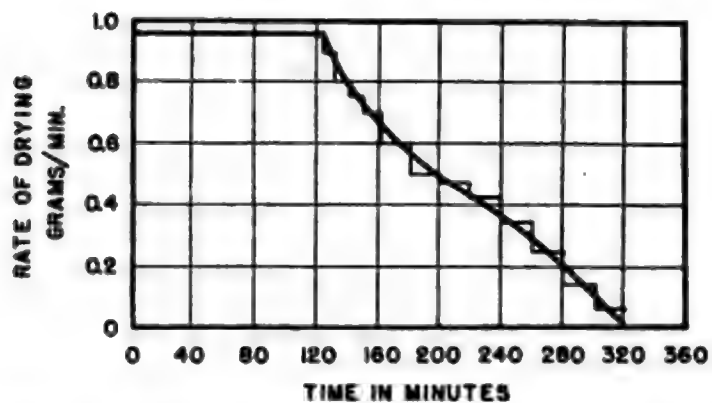


Figure 7. Drying Rate of Sawdust as a Function of Time

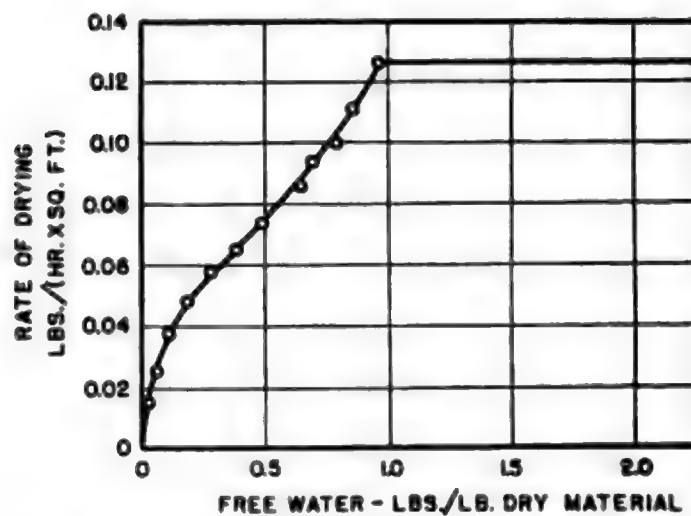


Figure 8. Drying Rate of Sawdust as a Function of Free Water Content

From the experimental data:

$$H_w - H_g = \frac{0.26}{1044} (131.5 - 85.5) = 0.0114 \text{ lb. water per lb. dry air}$$

where

131.5 = t_g = dry-bulb temp., °F. (average value from Columns 7 and 9 of Table 3).

85.5 = t_w = wet-bulb temp., °F. (average value from Columns 8 and 10 of Table 3).

1044 = λ_w = latent heat of evaporation of water at t_w , B.t.u. per lb. (from steam table).

Therefore, from Equation (2):

$$k_{a'} = \frac{0.127}{0.0114} = 11.1 \text{ lb. water evaporated per hr. per ft.}^2 \text{ per unit humidity difference.}$$

where

0.127 = lb. water evaporated per hr. per ft.² (from Figure 8).

Mass Velocity of the Air. This is calculated from the average velocity of the exit air, the average wet- and dry-bulb temperatures of the exit air, and the free cross-sectional area of the compartment with the pan in place. The steps in the calculation are:

1. *Humidity of the exit air.*

From Equation (3):

$$\begin{aligned} H_g &= H_w - \frac{0.26}{\lambda_w} (t_g - t_w) \\ &= 0.0335 - \frac{0.26}{1040.5} \times (162.7 - 92.3) \\ &= 0.0335 - 0.0176 = 0.0159 \text{ lb. water per lb. dry air.} \end{aligned}$$

where

162.7 = t_g = dry-bulb temp. °F. (average value from Column 5 of Table 3).

92.3 = t_w = wet-bulb temp. °F. (average value from Column 6 of Table 3).

1040.5 = λ_w = latent heat of water at t_w , B.t.u. per lb.

0.0335 = saturation humidity at t_w , lb. water per lb. dry air.

2. *Cubic feet of exit air per hour*

$$= 470.5 \times 60 \times (\pi \times 4^2/4) \times 1/144 = 2,466$$

where

470.5 = average velocity of the exit air, ft. per min. (from Column 11 of Table 3).

$(\pi \times 4^2/4) \times 1/144$ = cross-sectional area of the exit duct, ft.² (The diameter of the duct was 4 in.).

3. *Cubic feet of water vapor per cubic foot of dry air*
= mols of water vapor per mol of dry air

$$= \frac{H_g/18}{1/29} = 0.0159 \times 29/18 = 0.0255$$

where

H_g = humidity of the exit air, lb. water vapor per lb. dry air.

18 and 29 = molecular weight of water vapor and air, respectively.

4. *Cubic feet of dry air per cubic foot of exit air*
= $1/(1 + 0.0255) = 1/1.0255 = 0.975$

5. *Cubic feet of dry air per hour*
= (ft.³ exit air per hr.) \times (ft.³ dry air per ft.³ exit air)
= $2,466 \times 0.975 = 2,404$

6. *Pounds of dry air per hour*

$$= (\text{ft.}^3 \text{ dry air per hr.}) \times (\text{mol. wt.}) \times (1/359) \times \frac{492}{t_g + 460}$$

$$= 2,404 \times 29 \times 1/359 \times 492/(162.7 + 460) = 153.5$$

where

359 = cubic feet of gas per lb. mol at standard conditions.

$162.7 + 460$ = temp. of exit air, °R.

Therefore,

Mass velocity of the air over the pan

$$= 153.5/0.826 = 185.8 \text{ lb. dry air per hr. per ft.}^2 \text{ of free cross-sectional area.}$$

where

0.826 = free cross-sectional area (from Table 3).

Average Velocity of the Air Over the Pan. This is calculated from the mass velocity and the average humidity and temperature of the air over the pan.

Average humidity of the air over the pan

From Equation (3):

$$H_g = 0.0267 - \frac{0.26}{1044} (131.5 - 85.5) = 0.0152 \text{ lb. water per lb. dry air.}$$

where

131.5 = dry-bulb temp. of the air over the pan, °F. (average value from Columns 7 and 9 of Table 3).

85.5 = wet-bulb temp. of the air over the pan, °F. (average value from Columns 8 and 10 of Table 3).

0.0267 = saturation humidity of air at 85.5°F., lb. water per lb. dry air (from humidity chart).

1044 = latent heat of evaporation of water at 85.5°F., B.t.u. per lb. (from steam table.).

Therefore,

Average velocity of the air over the pan

$$\begin{aligned}
 &= [\text{lb. dry air}/(\text{hr.} \times \text{ft.}^2)] \times \frac{359}{3600} \times \frac{t_g + 460}{492} \times \left(\frac{1}{29} + \frac{H_g}{18} \right) \\
 &= 185.8 \times \frac{359}{3600} \times \frac{131.5 + 460}{492} \times \left(\frac{1}{29} + \frac{0.0152}{18} \right) \\
 &= 0.795 \text{ ft. per sec.}
 \end{aligned}$$

NOMENCLATURE

A = area of drying surface, ft.²

$\frac{dw}{d\theta}$ = drying rate, lb. water evaporated per hr.

H_g = humidity of air, lb. water per lb. dry air.

H_w = saturation humidity of air, lb. water per lb. dry air.

k_o = drying coefficient, lb. water evaporated per hr. per ft.² per unit humidity difference.

t_g = dry bulb temperature of the air, °F.

t_w = wet bulb temperature of the air, °F.

λ_w = latent heat of vaporization of water at t_w , B.t.u. per lb.

REFERENCES

1. BADGER AND McCABE, "Elements of Chemical Engineering," McGraw-Hill Book Company, New York, 2nd ed., 1936, p. 308.
2. Ibid., pp. 252-4.

A VACUUM TRAY DRIER

Designed and Constructed by
The Department of Chemical Engineering
University of North Dakota

Description. This drier, which is shown in Figure 1, consists of a steam-jacketed heating chamber and a condenser. The heating chamber is 2 feet long and is made from concentric 10- and 12-inch steel pipe; the total heating surface is 5.25 square feet.

The condenser, which is a length of 6-inch steel pipe fitted with a coil of $\frac{3}{8}$ -inch copper tubing, is attached directly to the bottom of the heating chamber in such a manner that it also serves as a base support.

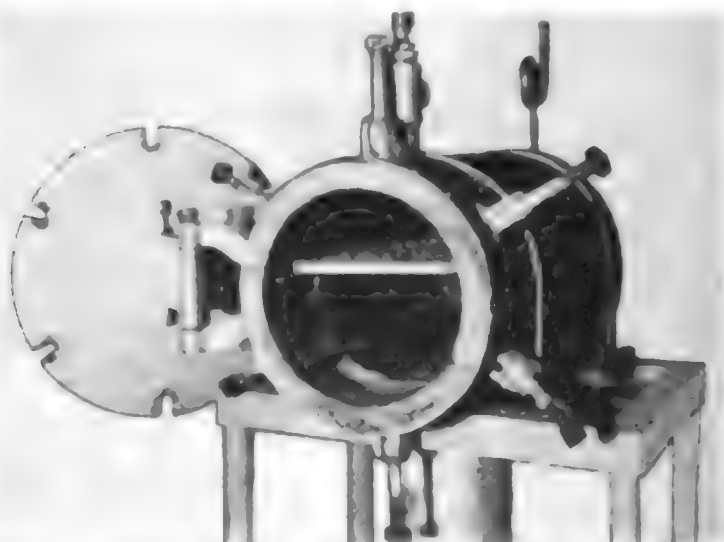


Figure 1. Vacuum Tray Drier—University of North Dakota

The material to be dried is placed in pans supported on angle iron racks that are welded to the inside of the drier. The drier, constructed as shown in Figure 2, has a total drying surface of 3 square feet, but additional racks can be installed if greater capacity is desired.

Materials and Cost. The materials used in the construction of this drier are given in Table 1. These materials cost about \$45, exclusive of the vacuum equipment.

Notes on Construction. This drier was constructed in the following manner:

1. The sections of 12- and 10-inch pipes for the drier shell and steam jacket were matched in length by facing the ends on a lathe, and the end plate *A* and ring *B* were turned to dimensions and welded in place.

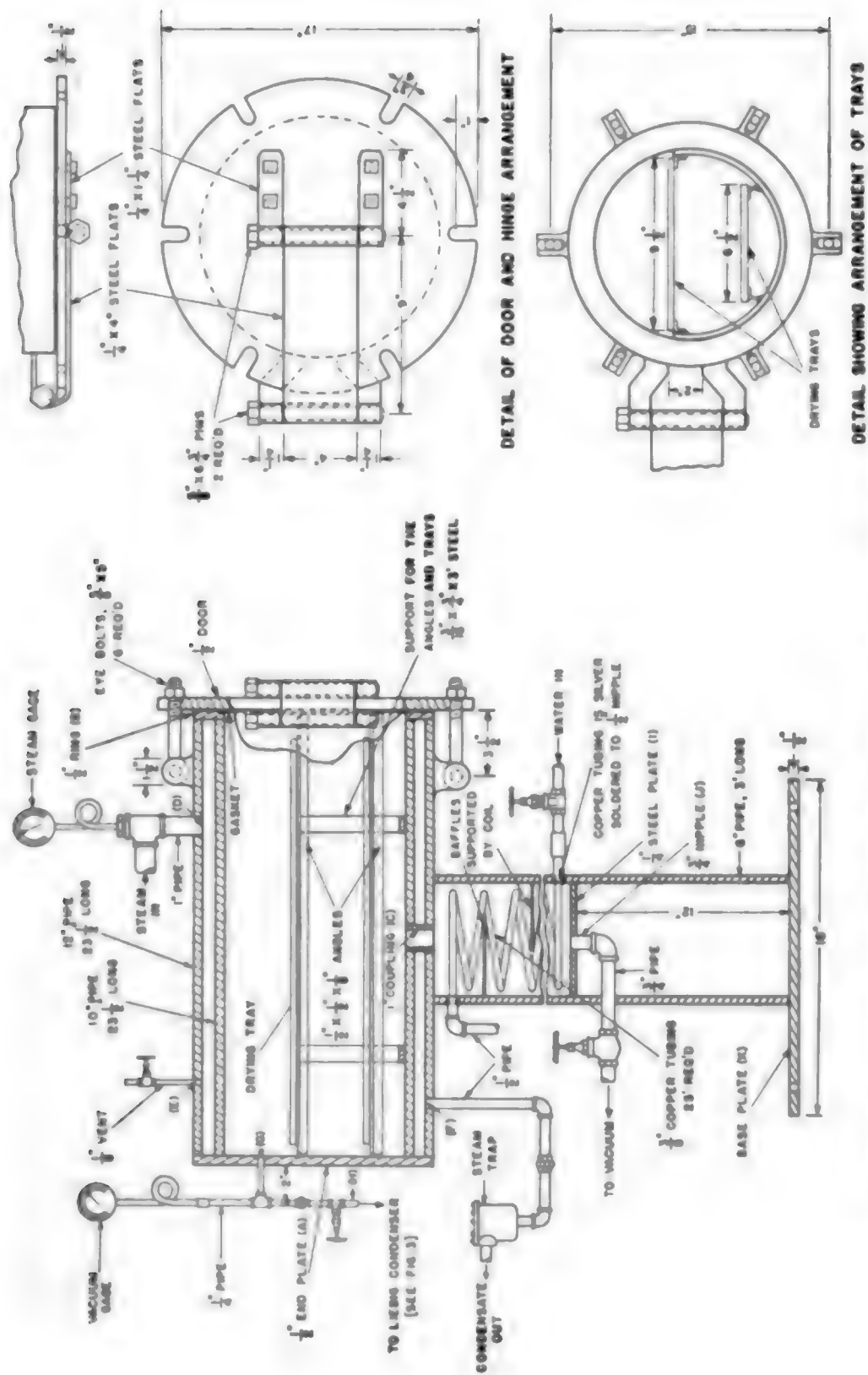


Figure 2. Assembly and Details of Vacuum Tray Drier—University of North Dakota

TABLE 1
Bill of Materials for Vacuum Drier
 University of North Dakota

Quantity	Description	Size	Material
2 ft.	Std. pipe	12 in.	Black wrought steel
2 ft.	" "	10 in.	" " "
3 ft.	" "	6 in.	" " "
1	Std. nipple	1 in. x 9 in. long	" " "
2	" "	1 in. x 3 in. "	" " "
1	" "	1 in. x 2½ in. "	" " "
1	" "	½ in., short	" " "
2	" "	½ in. x 4 in. long	" " "
1	" "	½ in. x 6 in. "	" " "
7	" "	½ in. x 2 in. "	" " "
3	" "	½ in. x 2 in. "	" " "
1	" "	½ in. x 3 in. "	" " "
1	Std. tee	1 in.	Black malleable iron
1	" "	½ in.	" " "
1	Std. 90° elbow	1 in.	" " "
1	" " "	½ in.	" " "
2	" " "	½ in.	" " "
1	Ground-joint union	1 in.	" " "
3	" " "	½ in.	" " "
1	Bushing	1 x ½ in.	Cast iron
1	Std. coupling	1 in.	Black malleable iron
1	Std. gate valve	1 in.	Brass
1	" " "	½ in.	"
1	" " "	½ in.	"
1	Std. globe valve	½ in.	"
1	" " "	½ in.	"
1	Steam trap	½ in.	Cast iron
1	Pressure gage	0 to 60 lb. press., 3½ in. face	Brass
1	Vacuum gage	30 in. vac., 3½ in. face	"
2	Steam-gage siphon	½ in.	Black wrought steel
25 ft.	Tubing	½ in.	Copper
8 ft.	Angles	½ x ½ x ½ in.	Steel
1½ ft.	Flats	4 x ½ in.	Cold-rolled steel
2 ft.	"	2 x ½ in.	" " "
2½ ft.	"	1½ x ½ in.	" " "
3 ft.	"	½ x ⅜ in.	" " "
1	Round plate	17 in. diam. x ½ in. thick	Steel
1	" "	12½ in. diam. x ½ in. thick	"

TABLE 1—*Concluded*

Quantity	Description	Size	Material
1	Round plate	6 in. diam. x $\frac{1}{2}$ in. thick	Steel
1	Ring	10 in. I.D. x $12\frac{1}{2}$ in. O.D. x $\frac{1}{2}$ in. thick	"
1	Plate	18 x 18 x $\frac{1}{2}$ in. thick	"
1	Bar	$\frac{1}{2}$ in. diam. x 14 in. long	Cold-rolled steel
1	Sheet	1 ft. x 1 ft. x 48 oz.	Copper
6	Eye bolts	$\frac{1}{2}$ in. diam. x 5 in. long	Steel
4	Machine bolts	$\frac{1}{2}$ in. — 13 NC x $\frac{1}{2}$ in. long	"
6	Square nuts	$\frac{1}{2}$ in. — 11 NC	"
2	Hexagonal nuts	$\frac{1}{2}$ in. — 11 NC	"
1	Gasket	10 in. I.D. x $12\frac{1}{2}$ in. O.D. x $\frac{1}{16}$ in. thick	Cranite
2 ft.	Pipe insulation	12 in.	85% magnesia

2. All the necessary holes were drilled in the inner and outer pipes; a 1-inch coupling was welded at *C*; and holes *D*, *E*, *F*, and *G* were tapped, and fitted with nipples to provide for the steam, air-vent, condensate, and vacuum lines. To insure tightness, the nipple at *G* was brazed to the end plate *A*.

3. The copper tubing for the condenser was coiled; baffles, cut from 48-ounce copper sheet, were soldered to it; and the coil was inserted into the condenser shell. The ends of the coil were inserted into steel nipples, which were welded to the condenser shell; the connections between the tubing and the nipples were made with silver solder. Steel plate *I*, nipple *J*, and base plate *K* were welded in place as shown, and the condenser was then welded to the body of the drier.

4. The door was finished to dimensions and clamped in position; the eye-bolt hinges were welded in place, and the hinge assembly was fabricated, attached to the door with screws, and welded to the body of the drier.

5. A gasket was cut to fit ring *B* and cemented in place; and the drier was then covered with insulation.

Comments on Design. To obtain accurate data when carrying out a drying experiment with rather small quantities of material, it is desirable to use a different condensing system because of the relatively large

hold-up in the copper-coil condenser. For such cases, opening *C* is stoppered, and a glass Liebig condenser is attached at *H* by means of rubber tubing. The end of the condenser is fitted with a glass adapter, which extends through a rubber stopper into a graduated cylinder; another connection is made from the top of the cylinder to a glass drying tower that rests on a small platform balance. The water which condenses is measured in the graduated cylinder, and that which does not

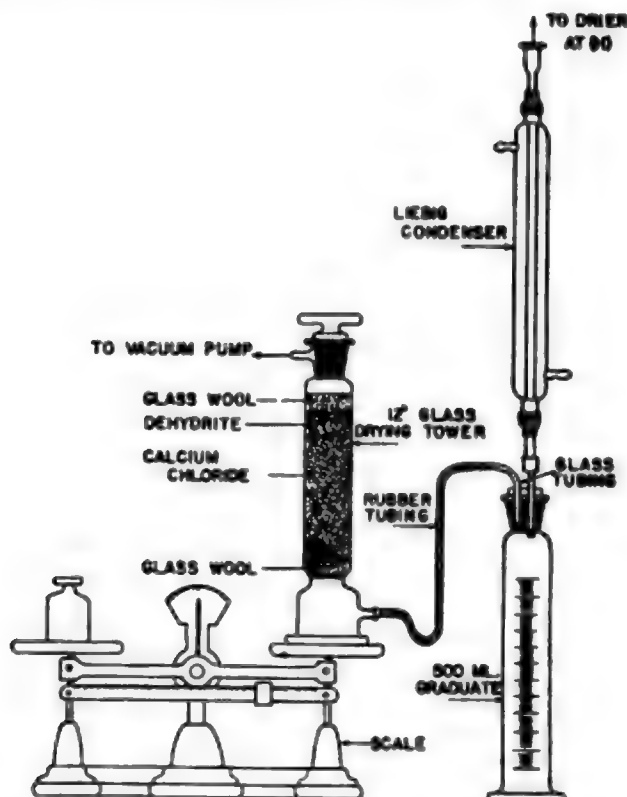


Figure 3. Moisture-Collecting System for Vacuum Drying Experiment

condense is absorbed in the drying tower and weighed. This setup is shown in Figure 3.

Operation. One or more trays are filled with the material that is to be dried, and placed in the drier. The drier is closed and, after adjusting the steam pressure to the desired value, the vacuum pump is started. Drying is continued until the sample is substantially dry; during this period, the quantity of water evaporated is determined periodically.

TYPICAL EXPERIMENT

Object. To study the drying characteristics of a small quantity of wet sawdust under vacuum.

Procedure. 1. Obtain the following equipment from the stock room and set it up as shown in Figure 3:

- a. A glass Liebig condenser and adapter.
 - b. A graduated cylinder of 500 ml. capacity.
 - c. A glass drying tower filled with calcium chloride and dehydrite, or other suitable drying agents.
 - d. A balance and weights.
2. Close opening *C* of the drier with a rubber stopper.
 3. Fill a pan with wet sawdust, measure its surface area, and place it in the drier.
 4. Close the drier, adjust the steam pressure to the desired value, and start the vacuum pump.
 5. Record the following readings at frequent intervals:
 - a. Pressure and vacuum.
 - b. Volume of condensate in the graduated receiver.
 - c. Weight of water absorbed in the drying tower.
 6. Continue drying until practically all of the water is removed from the material, and then remove and weigh the pan.
 7. Repeat the experiment with different steam pressures.

Data and Results. The drying experiment was carried out with wet sawdust in a single pan having a drying area of 1 square foot. Drying was carried out for 2 hours and 25 minutes, and at the end of this time the sawdust was substantially dry. The data obtained during this run are shown in Table 2.

The pan and its contents were weighed before the pan was placed in the drier and again at the end of the run; the weights were as follows:

	Grams
1. Empty pan	656
2. Pan + wet sawdust	956
3. Wet sawdust	300
4. Pan + dried sawdust	745
5. Dried sawdust	89
6. Initial moisture	211

Calculations. The experimental data were treated in the following manner:

1. The weight of water evaporated (Column 7 of Table 2) was plotted against elapsed time (Column 2 of Table 2); the curve is shown in Figure 4.

2. The curve of Figure 4 was graphically differentiated to determine the drying rates; the calculations involved in this differentiation are given in Columns 1 to 5 of Table 3, and the derived curve is plotted in Figure 5.

TABLE 2

Experimental Data for the Drying of Wet Sawdust Under Vacuum

Time, hr.: min.	Elapsed time, min.	Steam pressure, lb./in. ² gage	Vacuum, in. Hg	Vol. water collected, ml.	Wt. water absorbed, grams	Weight water evaporated, grams
1	2	3	4	5	6	7
7:05	0	14	0	0	0	0
7:06	1	14	4.5	0	0	0
7:07	2	14	7.0	1	0	1
7:08	3	13	10.0	1.5	0	1.5
7:11	6	12	13.5	2	0.4	2.4
7:13	8	12	16.5	3	0.6	3.6
7:15	10	12	18.5	4	0.8	4.8
7:19	14	11	20.0	5.5	1.0	6.5
7:20	15	11	21.0	8	1.2	9.2
7:22	16	12	21.8	10	1.5	11.5
7:23	18	13	22.2	13	1.8	14.8
7:25	20	15	23.0	18	1.8	19.8
7:26	21	16	23.2	21	2.0	23.0
7:27	22	18	23.5	24	2.2	26.2
7:29	24	17	24.0	30	2.4	32.4
7:31	26	13	24.0	36	2.7	38.7
7:33	28	11	24.1	41	3.0	44.0
7:35	30	11	24.2	45	3.2	47.2
7:37	32	14	24.2	47	3.5	50.5
7:39	34	16	24.3	53	3.7	56.7
7:40	35	17	24.3	55	3.7	58.7
7:41	36	18	24.3	58	4.0	62.0
7:44	39	17	24.3	63	4.2	67.2
7:48	43	16	24.3	72	4.5	76.5
7:57	52	11	24.4	96	6.0	102.0
8:03	58	17	24.5	109	6.5	115.5
8:09	64	15	24.5	129	7.2	136.2
8:15	70	16	24.5	138	8.0	146.0
8:21	76	18	24.5	151	8.5	159.5
8:30	85	15	24.5	165	9.5	174.5
8:35	90	12	24.6	171	10.2	182.2
8:43	98	12	24.6	180	11.0	191.0
8:47	102	12	24.8	183	11.6	194.6
8:50	105	12	24.8	184	12.0	196.0
8:55	110	12	24.8	188	12.6	200.6
9:00	115	12	24.8	190	13.2	203.2
9:05	120	12	24.8	191	14.0	205.0
9:10	125	12	24.6	193	14.5	207.5
9:15	130	12	24.6	193.5	15.0	203.5
9:20	135	12	24.6	193.5	15.5	209.0
9:25	140	12	24.6	193.8	16.0	209.8
9:30	145	12	24.6	193.8	16.7	210.5

3. Drying rates, read from the curve of Figure 5, are tabulated in Column 6 of Table 3. These values were converted to units of pounds

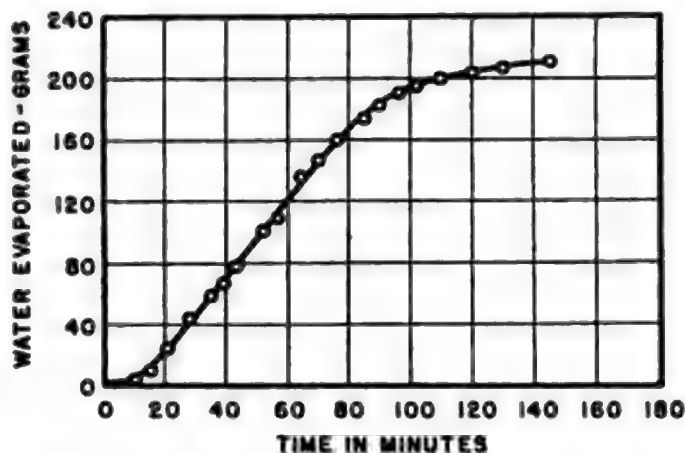


Figure 4. Loss of Moisture as a Function of Time

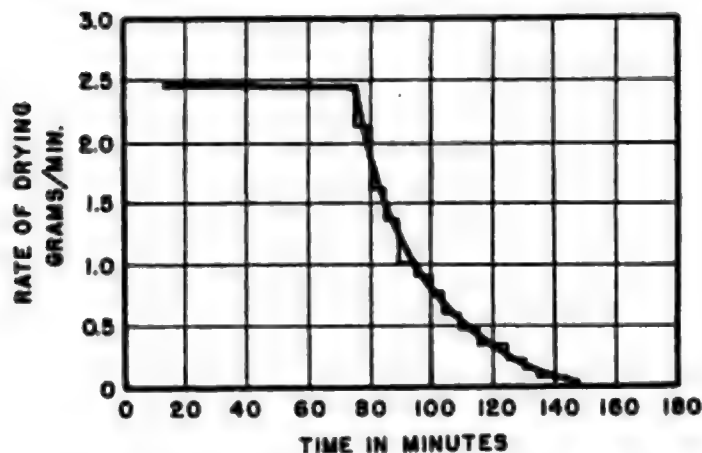


Figure 5. Drying Rate as a Function of Time

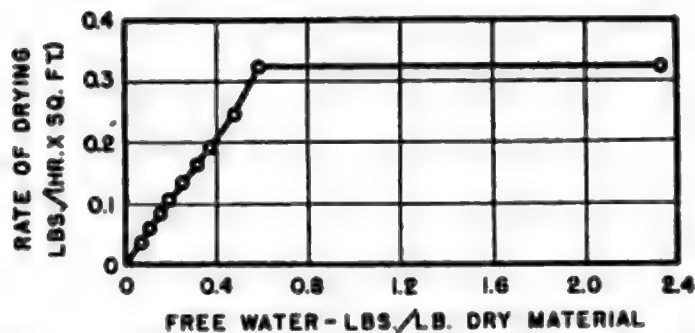


Figure 6. Drying Rate as a Function of Free Water Content

of water evaporated per square foot per hour by multiplying them by the factor:

$$\frac{\text{min./hr.}}{\text{grams/lb.}} \times \frac{1}{\text{pan area, ft.}^2} = \frac{60}{454} \times \frac{1}{1} = 0.0321$$

The drying rates in these units are given in Column 7 of Table 3.

4. The free water—the water remaining at time θ (Column 8 of Table 3)—was obtained by subtracting the total quantity of water evaporated up to time θ (Column 2 of Table 3) from the initial moisture content—211 grams.

TABLE 3

Summary of the Calculations of the Rate of Drying of Wet Sawdust under Vacuum

Elapsed time, θ , min. (From Fig. 4)	Water evaporated, w, grams. (From Fig. 4)	$\Delta\theta$, min.	Δw , grams	$\frac{\Delta w}{\Delta\theta}$, grams/min.	$\frac{dw}{d\theta}$, grams/min. (From Fig. 5)	Drying rate, lb./hr. \times ft. ³	Free water, grams	Free water, lb. water per lb. dry material
1	2	3	4	5	6	7	8	9
0		14	6	—				
14	6	62	154	2.483	2.483	0.3281	205	2.302
76	160	4	9	2.25	2.483	0.3281	51	0.573
80	169	5	8	1.6	1.84	0.2431	42	0.472
85	177	5	7	1.4	1.46	0.1930	34	0.382
90	184	5	5	1.0	1.20	0.1585	27	0.303
95	189	5	4.5	0.9	1.00	0.1321	22	0.247
100	193.5	5	4	0.8	0.80	0.1057	17.5	0.1966
105	197.5	5	3	0.6	0.63	0.0832	13.5	0.1515
110	200.5	5	2.5	0.5	0.49	0.0649	10.5	0.1180
115	202	5	2	0.4	0.40	0.0527	9.0	0.1011
120	204	5	2	0.4	0.30	0.0396	7.0	0.0786
125	206	5	2	0.4	0.22	0.0291	5.0	0.0562
130	208	5	1	0.2	0.16	0.0211	3.0	0.0337
135	209	5	0.5	0.1	0.11	0.0145	2.0	0.0225
140	209.5	5	0.5	0.1	0.07	0.0092	1.5	0.0169
145	210				0.04	0.0053	1.0	0.0112

5. The free water content in lb. water per lb. dry sawdust (Column 9 of Table 3) was calculated by dividing the weight of water remaining at time θ by the weight of dried sawdust—89 grams.

6. The drying rates in lb./hr. \times ft.³ were then plotted against free water content; this curve is shown in Figure 6.

A VACUUM DRIER MODIFIED FOR EXPERIMENTAL PURPOSES

By

The Department of Chemical Engineering
University of Florida*

Description. This unit, shown in Figure 1, is a *Bufflovac* single-tray drier which has been modified to permit the determination of drying rates without the necessity for removing the material from the drier.

The system is arranged in such a manner that the water evaporated in the drying compartment is condensed in a copper coil and discharged into a funnel from which it flows to a receiver. Drying is aided by passing a stream of air through the system; the air is admitted through the door of the drier after first passing through an orifice meter. Measurement of the rate of flow of the air is necessary so that the quantity of water it picks up can be calculated.

Materials and Cost. The materials needed to make the necessary modifications of the drier are listed in Table 1. The total cost of these items is approximately \$10.

Notes on Construction. 1. The drier was opened on the plane *A-A*, and the copper funnel *F*, soldered to a short section of $\frac{1}{2}$ -inch brass pipe, was placed in position so as to collect the condensate from the cooling coil and deliver it to the receiver *D*, which was made from a 1-foot length of 3-inch brass pipe and a water gage.

2. The orifice *E* was made by drilling a small opening in a thin brass plate, about $\frac{1}{8}$ -inch O.D., and soldering this plate to the end of a $\frac{1}{8}$ -inch I.P.S. brass nipple provided with a single lead connected to an open-end water manometer by means of rubber tubing.

Operation. Steam is admitted to the drier and adjusted to the desired pressure; and then one or more sheets of the material to be dried are supported in the drier so as to expose their entire surfaces; the door is closed; the vacuum pump is turned on; and needle valve *G* is adjusted to admit the proper amount of air to maintain the desired vacuum.

Condensate is collected in receiver *D* with valve *B* open and valve *C* closed; at intervals, valve *B* is closed, and the condensate is run out through valve *C*, and weighed.

* Reported by W. H. Beisler.

TYPICAL EXPERIMENT

Object. To study the rate of drying of wet fiber insulating board in a vacuum drier.

Procedure. 1. Place a weighed sheet of wet fiber board in the drier.

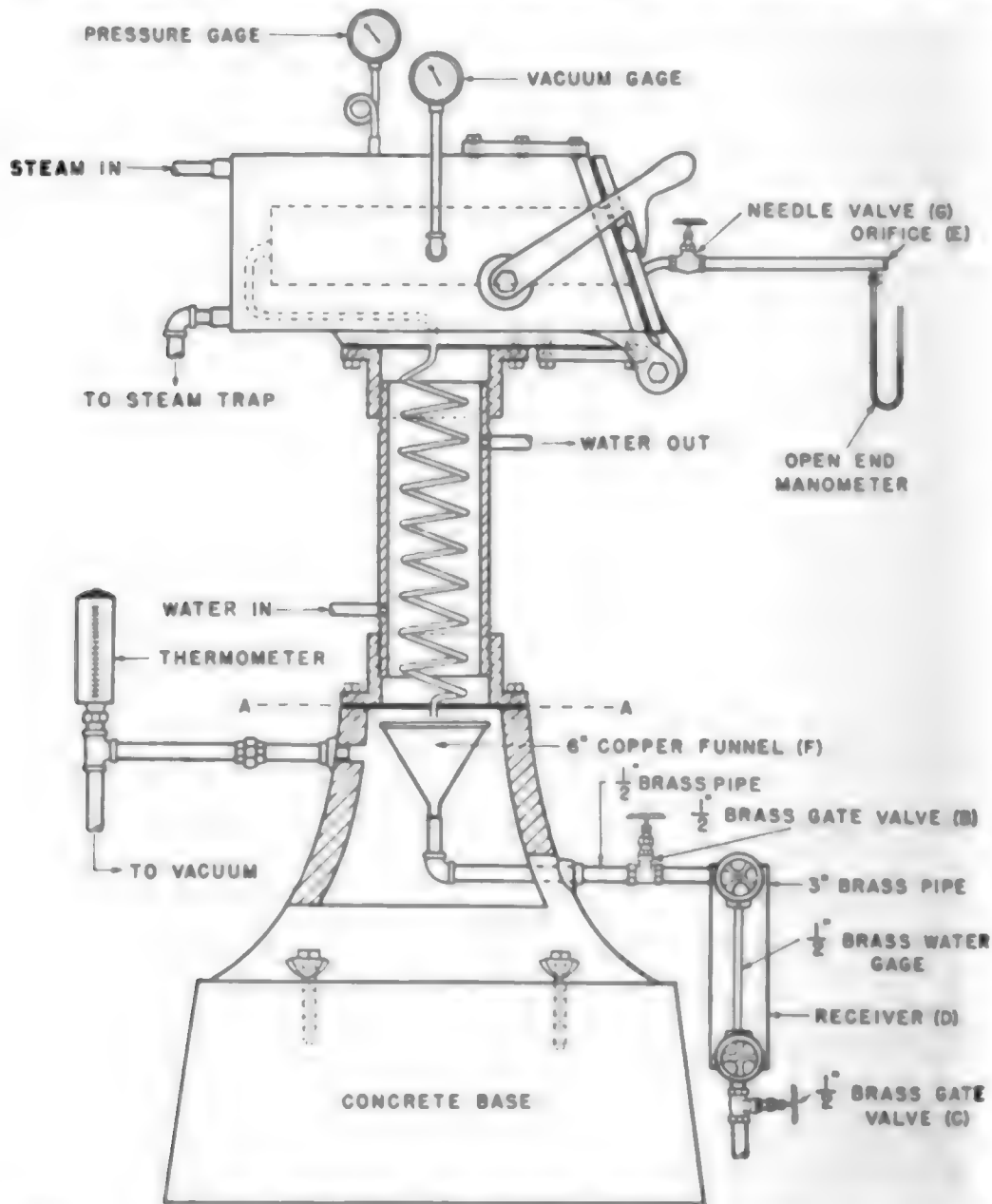


Figure 1. Vacuum Drier Modified for Experimental Purposes—University of Florida

and dry it at about 20 pounds per square inch gage steam pressure and with a vacuum of about 10 inches of mercury. Continue the drying until the stock is substantially dry, maintaining the steam pressure and vacuum constant throughout the experiment.

2. Make a duplicate run under the same conditions and then repeat the experiment at a different vacuum.
3. During each run, record the following data at frequent intervals:
 - a. Wet and dry bulb temperatures of the inlet air.
 - b. Temperature of the exit air.
 - c. Orifice manometer reading.
 - d. Steam pressure and vacuum.
 - e. Weight of condensate.
4. At the end of the experiment, weigh the dried stock and measure its area.

TABLE 1
Bill of Materials for Modification of Vacuum Drier
 University of Florida

Quantity	Description	Size	Material
	<i>For Orifice Assembly</i>		
3 ft.	Std. pipe	$\frac{1}{2}$ in.	Galv. wrought steel
2	Std. elbow	$\frac{1}{2}$ in.	Galv. malleable iron
1	Ground-joint union	$\frac{1}{2}$ in.	" " "
1	Round plate	$\frac{1}{2}$ in. O.D. x $\frac{1}{2}$ in. thick	Brass
1	Needle valve	$\frac{1}{2}$ in.	"
	<i>For Funnel Assembly</i>		
1 ft.	Std. pipe	3 in.	Brass
1 $\frac{1}{2}$ ft.	" "	$\frac{1}{2}$ in.	"
1	Elbow	$\frac{1}{2}$ in.	"
1	Funnel	6 in.	Copper
1	Water gage	$\frac{1}{2}$ in. with 10 in. glass	Brass
2	Std. gate valve	$\frac{1}{2}$ in.	"

5. From the experimental results, plot the following:
 - a. Grams of water evaporated against elapsed time in minutes.
 - b. Grams of water evaporated per minute against elapsed time in minutes.
 - c. Pounds of water evaporated per square foot per hour against pounds of free water per pound of dry stock.
6. Calculate:
 - a. The rate of drying during the constant rate period.
 - b. The value of the constant k_d in the constant-rate equation.
 - c. The drying time theoretically required. Compare this value with the observed drying time.

Experimental Data and Results. The experimental data and calcu-

TABLE 2

Experimental and Calculated Results for Vacuum Drying of Fiber Board

Material: Fiber insulation board

Drying area: 12.67 ft.²

Wet weight: 4.170 kg. = 9.17 lb.

Dry weight: 1.909 kg. = 4.2 lb.

Total water evaporated: 2.261 kg. = 4.97 lb.

Steam Pressure: 20 lb./in.² gage

Vacuum: 10 in. Hg

Elapsed time, hr.	Inlet air temp., °F.		Orifice manometer, in. water	Exit air temp., °F.	Water condensed, grams	Water condensed plus water picked up by air, grams	Total water removed, grams (cumulative)	Drying rate, grams/min.	Drying rate, lb./hr. × ft. ²	Free moisture, lb. water lb. dry stock
	dry-bulb	wet-bulb								
1	2	3	4	5	6	7	8	9	10	11
0.00					87.7	115		3.82*		1.185
0.50	73.6	63.5	13.00	72	191.3	218.5	115	7.28	0.077	1.125
1.00	75.0	63.7	13.06	73	184.2	211.5	333.5	7.05	0.075	1.010
1.50	76.5	64.0	12.70	74	165.6	192.9	545.0	6.44	0.070	0.898
2.00	77.7	64.2	12.00	74.5	159.5	186.8	737.9	6.22	0.066	0.798
2.50	77.4	63.5	11.75	75	147.0	174.3	924.7	5.81	0.062	0.700
3.00	77.0	63.2	11.60	75	134.6	161.9	1099.0	5.40	0.058	0.608
3.50	77.0	63.0	11.67	75	114.5	141.8	1260.9	4.72	0.053	0.524
4.00	76.0	61.5	11.00	75	196.2	250.7	1402.7	4.18	0.049	0.450
5.00	77.0	61.0	10.85	75	146.9	201.4	1653.4	3.36	0.041	0.319
6.00	79.3	61.4	10.85	76	98.8	153.3	1854.8	2.56	0.033	0.213
7.00	79.0	61.2	10.85	76	65.2	119.7	2008.1	1.99	0.024	0.133
8.00	77.7	61.4	10.80	77	23.1	77.6	2127.8	1.30	0.016	0.069
9.00	77.4	61.2	10.75	77	3.8	31.1	2205.4	1.03	0.008	0.029
9.50	77.0	61.5	10.70	77			2236.5			0.013
Av....	77.0	62.3	11.54	75						

* This value is low because conditions had not yet become constant.

lated results for the drying of insulating fiber board in the modified vacuum drier are given in Table 2. These data were treated in the following manner:

1. The weights of water evaporated during the experiment (Column 8) were plotted against time (Column 1), as shown in Figure 2, and this

curve was graphically differentiated to obtain the drying rate curve, Figure 3. This was done most conveniently by dividing the values in

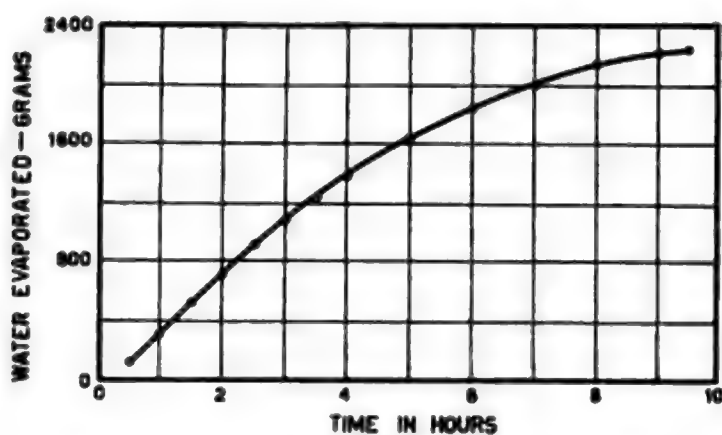


Figure 2. Loss of Moisture as a Function of Time

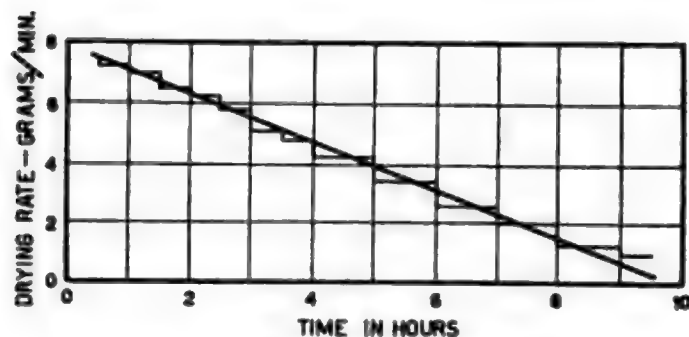


Figure 3. Drying Rate as a Function of Time

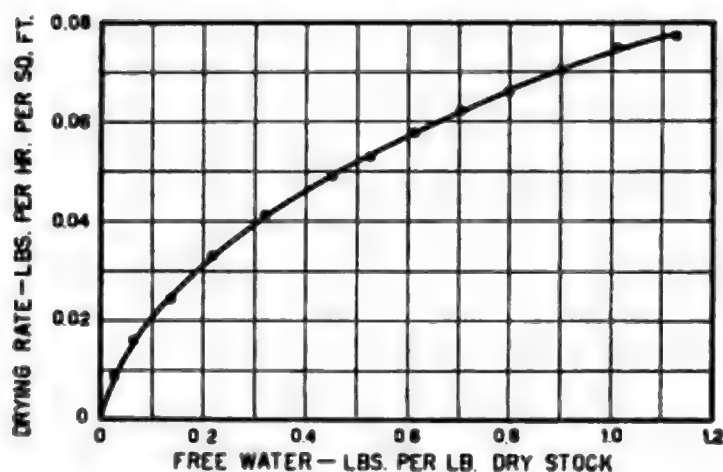


Figure 4. Drying Rate as a Function of Free Moisture Content

Column 7 by the corresponding time intervals in minutes, and then plotting the resulting values (Column 9) against time (Column 1) and drawing the best curve, which in this case was a straight line.

2. Drying rates, read from the curve of Figure 3, were converted into the units of lb./hr. \times ft.³ and plotted, in the usual manner, against the free moisture content to obtain the curve of Figure 4; the values used in plotting this curve are given in Columns 10 and 11.

Sample Calculations.

1. *Humidity of inlet air, H_1 .*

Average dry-bulb temp. = 77°F.

Average wet-bulb temp. = 62.3°F.

$H_1 = 0.0084$ lb. water/lb. dry air. (from humidity chart)

2. *Humidity of exit air, H_2 .*

The air leaving the drier was saturated with water vapor at 75°F. and at a pressure of $29.92 - 10 = 19.92$ in. (507 mm.) of mercury.

The vapor pressure of water at 75°F. = 22.2 mm. of mercury.

The partial pressure of the dry air = $507 - 22.2 = 484.8$ mm. of mercury.

$$H_2 = \frac{22.2 \times 18}{484.8 \times 29} = 0.0285 \text{ lb. water/lb dry air}$$

3. *Pounds of dry air per hour.*

Orifice manometer reading = 11.54 in. of water, which is equivalent to 79 ft.³ of entering air per hr. (from orifice calibration).

Average molecular weight of air = 29

Molecular weight of water vapor = 18

Therefore,

$$\frac{79}{1 + \frac{29}{18} \times H_1} = \frac{79}{1 + \frac{29}{18} \times 0.0084} = \frac{79}{1 + 0.014} = 78 \text{ ft.}^3 \text{ dry air/hr.}$$

and

$$\frac{78}{359} \times \frac{492}{460 + 77} \times 29 = 5.8 \text{ lb. dry air/hr.}$$

4. *Water picked up by air*

= $(H_2 - H_1) \times \text{lb. dry air/hr.} = (0.0285 - 0.0084) \times 5.8$

= 0.12 lb./hr. = $0.12 \times 454 = 54.5$ grams/hr.

5. *Water evaporated during measured interval of time* (Column 7)

= grams water condensed (Column 6) + grams water removed by air.

For example, for the period 0.5 to 1.0 hr.,

grams water condensed = 191.3

grams water removed by air = $(1.0 - 0.5) \times 54.5 = 27.2$

Therefore,

water evaporated during this period = $191.3 + 27.2 = 218.5$ grams.

6. *Drying rate in lb. per hr. per ft.²* (Column 10).

The values in this column were obtained by multiplying the drying rates in grams per minute (Figure 3) by the factor:

$$\frac{\text{min.}}{\text{hr.}} \times \frac{\text{lb.}}{\text{gram}} \times \frac{1}{\text{area in ft.}^2} = 60 \times \frac{1}{454} \times \frac{1}{12.67} = 0.0105$$

For example, when the elapsed time was 0.50 hr., the drying rate was

$$7.3 \times 0.0105 = 0.077 \text{ lb.}/(\text{hr.} \times \text{ft.}^2)$$

7. *Free moisture content, F* (Column 11).

Total moisture in sample = 2261 grams (from initial and final weights).

Weight of dried stock = 1909 grams (from final weight).

Therefore,

$$F = \frac{2261 - \text{water removed, (cumulative)}}{1909}$$

For example, when the elapsed time was 1 hr.,

$$F = \frac{2261 - 333.5}{1909} = 1.010 \text{ grams/gram or lb./lb.}$$

Theoretical Drying Time. To compare the observed drying time with the time theoretically required, a calculation of the theoretical drying time between free moisture contents of 1.125 lb. water per lb. dry stock and 0.069 lb. water per lb. dry stock was made using the equation for the falling rate period¹:

$$\theta = \frac{D}{A} \int_{F_1}^{F_2} \frac{dF}{f(F)} \quad (1)$$

where

θ = theoretical drying time, hr.

D = weight of dry stock, lb. = 4.2

A = drying area, ft.² = 12.67

F_1 = free moisture content at start of drying period, lb. water/lb. dry stock
= 1.125

F_2 = free moisture content at end of drying period, lb. water/lb. dry stock
= 0.069

$f(F)$ = rate of drying at free moisture content F , lb./(\text{hr.} \times ft.²).

As the drying-rate curve (Figure 4) was not a straight line, it was necessary to determine the value of the integral graphically. This

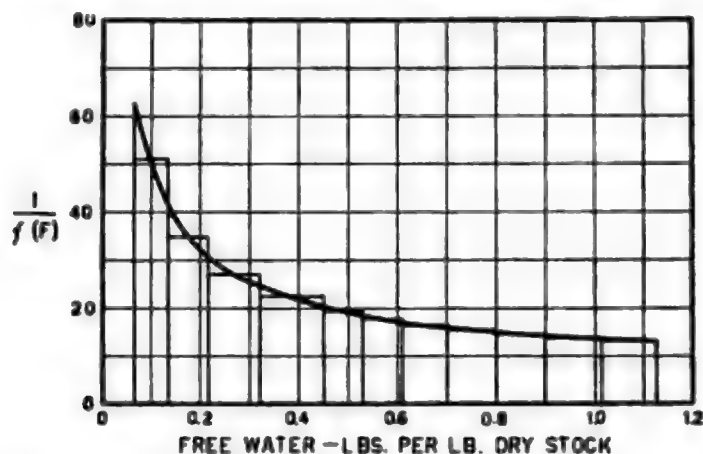


Figure 5. Graphical Integration of Drying Equation

TABLE 3
Data for Graphical Integration of Drying Equation

Free moisture content, F , lb. water/lb. dry stock	Drying rate, $f(F)$, lb. water/ (hr. \times ft. ²)	$\frac{1}{f(F)}$	ΔF	$\left[\frac{1}{f(F)}\right]_{\text{mean}}$	$\Delta F \left[\frac{1}{f(F)}\right]_{\text{mean}}$
1.125	0.077	13.0			
1.010	0.075	13.3	0.115	13.2	1.52
0.898	0.070	14.3	0.112	13.8	1.55
0.798	0.066	15.2	0.100	14.8	1.48
0.700	0.062	16.1	0.098	15.6	1.53
0.608	0.058	17.2	0.092	16.6	1.53
0.524	0.053	18.9	0.084	18.0	1.51
0.450	0.049	20.4	0.074	19.5	1.45
0.319	0.041	24.4	0.131	22.2	2.66
0.213	0.033	30.3	0.106	27.0	2.86
0.133	0.024	41.6	0.080	35.0	2.80
0.069	0.016	62.6	0.064	51.0	3.25

$$\text{Area under the curve} = \int_{0.069}^{1.125} \frac{dF}{f(F)} = \Sigma \Delta F \left[\frac{1}{f(F)} \right]_{\text{mean}} = 22.14$$

was done by plotting the reciprocal of the drying rate (Column 10) against the free moisture content (Column 11), as shown in Figure 5.

and determining the area beneath the curve between the limits of $F = 1.125$ and $F = 0.069$. The values used in plotting this curve and the results of the integration are given in Table 3. The value of the integral is 22.14; therefore

$$\theta = \frac{4.2}{12.67} \times 22.14 = 7.35 \text{ hr.}$$

Comments on Results. It is evident from Figures 3 and 4 that the initial moisture content of the stock was below the critical moisture content, and drying took place only during the falling-rate period. Therefore, the drying coefficient k_d could not be determined. It would have been better if the moisture content of the stock had been above the critical value so that the drying characteristics during the constant-rate period could have been studied.

The drying time calculated from Equation (1) is in very good agreement with the experimental time: the two values are 7.35 and 7.5 hours, respectively.

NOMENCLATURE

A = drying area, ft.²

D = weight of dry stock, lb.

$f(F)$ = rate of drying at free moisture content F , lb./hr. \times ft.².

F = free moisture content, lb. water/lb. dry stock.

H = humidity, lb. water/lb. dry air.

θ = drying time, hr.

Subscripts

1 = initial condition.

2 = final condition.

REFERENCE

1. BADGER AND McCABE, "Elements of Chemical Engineering," McGraw-Hill Book Company, New York, 2nd ed., 1936, p. 309.

A CONTINUOUS COUNTERCURRENT TUNNEL DRIER

Designed and Constructed by
The Department of Chemical Engineering
Montana State College*

Description. This inexpensive drier, which provides a drying surface of about 9.36 square feet, is made of wood and Masonite presdwood. It consists of a tunnel, approximately $18\frac{1}{2}$ feet long by 0.2 square foot in cross-sectional area, and a baffled chamber which directs the drying air through the tunnel and permits the removal of the dried product with minimum disturbance to the drying conditions. The stock is dried in pans, which are pushed intermittently through the tunnel counter-current to the warm air, supplied by a Grinnell cabinet drier.**

The details of construction of this drier are shown in Figure 1. The dimensions given here are for a drier designed to accommodate pans $11\frac{1}{2} \times 7\frac{1}{2} \times 1\frac{1}{2}$ -inches. If pans of a different size are to be used, the dimensions of the drier should be modified accordingly.

Materials and Costs. The materials used in the construction of this drier are listed in Table 1; their total cost is only about \$13. However, the following additional equipment is needed to operate the drier experimentally:

1. Twenty thermometers: 0–220° F.
2. Ten rubber stoppers: No. 9.
3. Ten rubber stoppers: No. 0.
4. Ten wicks.
5. Air heater and blower.
6. Scale or balance: about 500-gram capacity.

Operation. *For Tests Under Constant Drying Conditions.* A pan filled with the wet stock is placed in the tunnel near the product entrance end between two pair of wet- and dry-bulb thermometers. The air velocity and temperature are adjusted to the desired values, and the pan is removed and weighed periodically until the stock reaches its equilibrium weight. Before the pan is removed from the drier for a weighing, readings are made of both pair of thermometers; humidity calculations are based on the averages of the readings.

For Tests Under Variable Drying Conditions. 1. Pans of wet stock

* Reported by J. L. Beal.

** A tempering coil such as shown in Figure 2 page 176 is recommended where an auxiliary drier is not available.

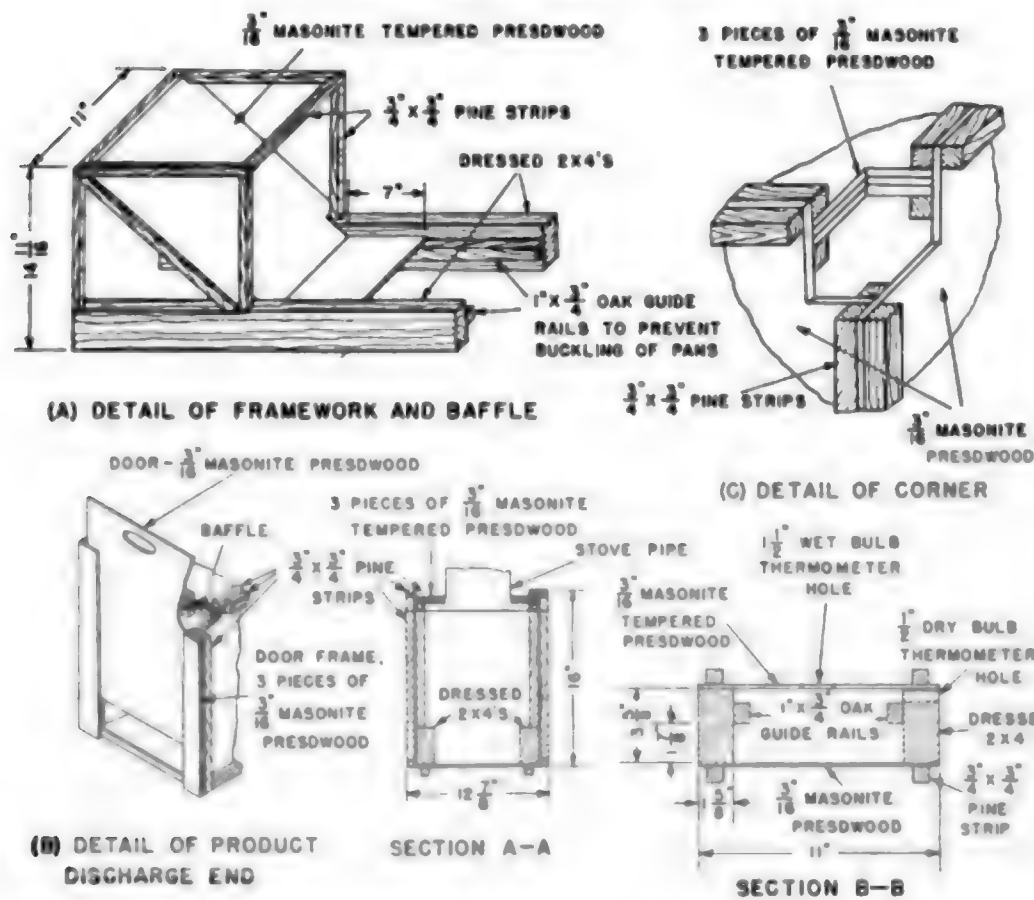
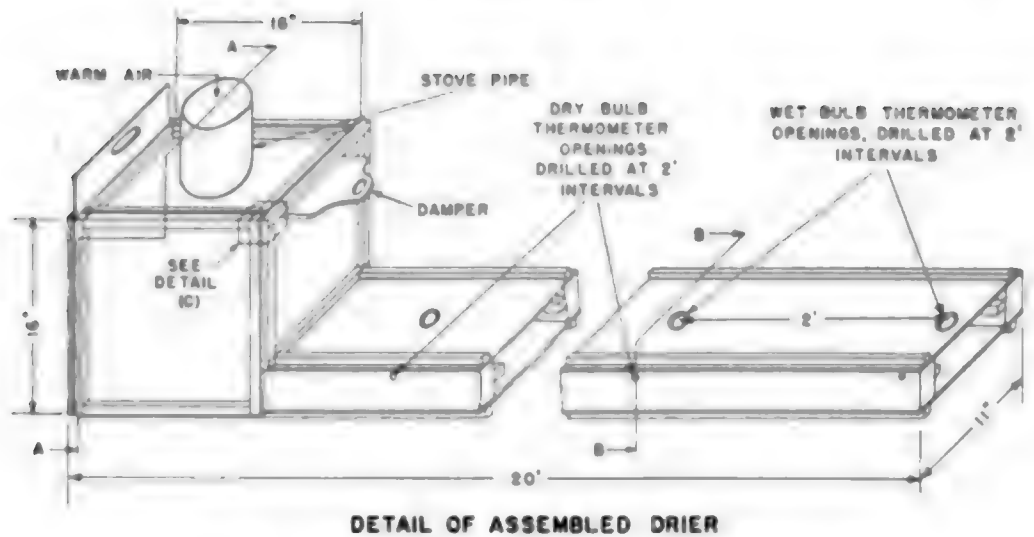


Figure 1. Assembly and Details of Continuous Countercurrent Tunnel Drier—University of Montana

are pushed in intermittently at constant intervals of a few minutes. After the tunnel has become filled with pans, a pan of dried stock is removed from one end of the drier at the same time that a pan of wet

stock is introduced at the opposite end. The drier holds 20 pans and, at the rate of one pan every 4 minutes, the drying time is 80 minutes. Each pan is weighed before and after it passes through the drier.

2. During the run, the temperature and the velocity of the entering air are maintained at the desired conditions, and all of the wet- and dry-bulb thermometers are read at regular intervals.

3. Drying is continued for about one hour after substantially steady conditions have been reached.

TYPICAL EXPERIMENT

Object. To study the drying characteristics of wet filter-cel under constant and variable conditions.

TABLE 1
Bill of Materials for Tunnel Drier
Montana State College

Quantity	Description	Size	Material
2 pieces	Lumber	2 x 4's, 20 ft. long	Pine; No. 1 common
100 ft	Wood strips	$\frac{1}{2}$ x $\frac{1}{2}$ in.	Pine
40 ft.	Lumber	1 x $\frac{1}{2}$ in.	Oak
44 sq. ft.	Masonite presdwood	$\frac{1}{8}$ in. thick	—
21 sq. ft.	Masonite tempered presdwood	$\frac{1}{8}$ in. thick	—
4	Brackets	15 x 15 in.	Steel
36	Pans	11 $\frac{1}{2}$ x 7 $\frac{1}{2}$ x 1 $\frac{1}{2}$ in.	Tin-plated steel

Procedure. *Under Constant Drying Conditions.*

1. Pass heated air from the cabinet drier through the tunnel drier, and adjust the damper so that the temperature of the air entering the tunnel drier is about 140°F.

2. Fill a tared pan to a depth of about $\frac{1}{2}$ -inch with wet filter-cel containing about 75 per cent water, weigh the pan and contents, and measure the exposed surface area.

3. Place the pan in the tunnel between two pair of wet- and dry-bulb thermometers near the product entrance end, and read the thermometers and weigh the pan at intervals until there is no further loss in weight.

4. Repeat this procedure with one or two additional pans of stock.

5. For each run, calculate the drying coefficient k_a in the following manner:

a. Plot the weight of the sample of stock against time, and from the slope of the curve compute the drying rate as a function of the free moisture content.

b. From the wet- and dry-bulb thermometer readings, determine the average humidity and temperature of the air during the constant-rate period.

c. Calculate the drying coefficient during the constant-rate period by means of the equation¹:

$$\frac{dw}{d\theta} = k_a (H_w - H_a) A \quad (1)$$

*Under Variable Drying Conditions**

1. Adjust the temperature and the velocity of the air to the same conditions as for the experiment under constant drying conditions.

2. Fill about 25 pans with the wet filter-cel, and determine the average exposed surface area. The moisture content of the wet stock and the depth to which the pans are filled should be about the same as in the experiment under constant-drying conditions.

3. Introduce several pans into the tunnel, and then add additional pans one at a time at intervals of four minutes. Weigh each pan just before it is introduced into the drier and, after the drier becomes filled, weigh each pan as it is pushed out by an incoming pan.

4. Before introducing a fresh pan, read all of the wet- and dry-bulb thermometers along the tunnel.

5. Continue drying for at least one-half hour after the thermometer readings and the loss in weight per pan have become substantially constant.

6. From these data, calculate the drying coefficient as follows:

a. Determine the average loss of moisture per pan over the period during which conditions were substantially constant.

b. Average the readings of the wet- and dry-bulb thermometers at each of the several stations; calculate the value of the humidity difference ($H_w - H_a$) at each thermometer station, and plot these values against the length of the drier; and calculate the mean humidity difference in the drier by integrating this curve and dividing the value of the integral by the length of the drier.

c. Calculate the drying coefficient k_a by means of Equation (1).

Experimental Data and Calculations for Drying Under Variable Conditions. The experimental data for one run under variable drying conditions are given in Tables 2 and 3, and the temperature data of Table 3 are also shown graphically in Figure 2.

* This experiment is based upon drying only above the critical moisture content. The experimental procedure is the same when drying is carried out below the critical moisture content but the calculations are more difficult.²

TABLE 2

Experimental Data for the Drying of Filter-cel in a Tunnel Drier

- (1) Total weight of wet stock = 8,153 grams
 (2) Moisture content of wet stock = 70.7 per cent
 (3) Weight of wet stock per pan = 216.5 grams
 (4) Depth of wet stock in pan = $\frac{1}{2}$ in.
 (5) Drying area per pan = 0.468 ft.²
 (6) Number of pans in drier = 20
 (7) Free cross-sectional area of tunnel with pans in place = 0.108 ft.²

Pan No.	Time, min.		Drying time, min.	Wt. pan plus sample, grams		Loss in wt., grams
	in	out		in	out	
1	0	64	—	435.3	387.0	48.3
2	0	68	—	434.2	392.0	42.2
3	0	72	—	445.5	407.5	38.0
4	0	76	—	437.7	403.5	34.2
5	0	80	—	453.0	416.0	37.0
6	4	84	80	442.0	406.0	36.0
7	8	88	80	438.0	405.5	32.5
8	12	92	80	442.5	406.5	36.0
9	16	96	80	441.5	406.5	35.0
10	20	100	80	436.7	401.0	35.7
11	24	104	80	441.5	409.0	32.5
12	28	108	80	436.5	406.0	30.5
13	32	112	80	440.7	409.0	31.7
14	36	116	80	433.0	403.0	30.0
15	40	120	80	435.0	403.0	32.0
16	44	124	80	436.0	407.0	29.0
17	48	128	80	437.5	407.0	30.5
18	52	132	80	443.5	413.0	30.5
19	56	136	80	450.0	418.5	31.5
20	60	140	80	451.5	423.0	28.5
21	64	144	80	442.0	412.5	29.5
22	68	148	80	450.5	419.0	31.5
23	72	152	80	437.5	404.5	33.0
24	76	156	80	437.7	406.0	31.7
25	80	160	80	434.0	402.5	31.5

The calculations, which are illustrated below, are based upon the interval between 100 and 136 minutes because the conditions were relatively constant during this period.

Calculation of the Drying Coefficient. When drying is carried out above the critical moisture content, the drying coefficient k_a' can be calculated from Equation (1)

$$\frac{dw}{d\theta} = k_a' (H_w - H_a) A \quad (1)$$

TABLE 3

Average Temperature Readings During Period of 100 to 136 Minutes

	Station No.					
	1	2	3	4	5	6
Dry-bulb temp., °F.....	81.3	86.6	—	104.3	120.1	138.9
Wet-bulb temp., °F.....	71.7	71.7	72.6	73.6	73.7	79.0

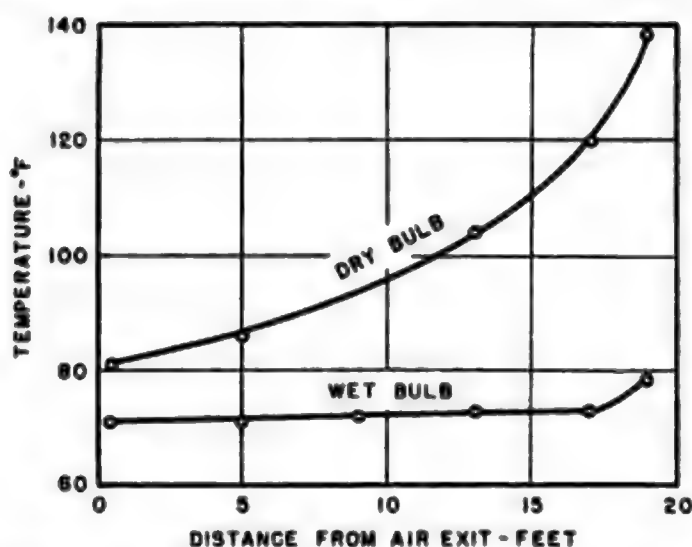


Figure 2. Wet- and Dry-Bulb Temperatures in Tunnel Drier

where

$\frac{dw}{d\theta}$ = average rate of drying, lb. water/hr.

H_w = humidity of air saturated at the temperature of the stock, lb. water/lb. dry air.

H_a = humidity of the air, lb. water/lb. dry air.

A = area of drying surface, ft.²

k_a' = drying coefficient, lb. water evaporated per hr. per ft.² per unit humidity difference.

1. **Drying Rate.** This is calculated from the average loss of moisture per pan and the number of pans passing through the drier per hour.

Each pan lost an average of 31 grams of water (from Table 2), and the pans were introduced at 4 minute intervals or at the rate of 15 pans per hour.

Therefore,

$$\frac{dw}{d\theta} = \frac{31}{454} \times 15 = 1.023 \text{ lb./hr.}$$

2. *Area of Drying Surface.* This is equal to the product of the drying area per pan—0.468 ft.²—and the number of pans in the drier—20.

Therefore,

$$A = 20 \times 0.468 = 9.36 \text{ ft.}^2$$

3. *Mean Humidity Difference.* The humidity difference ($H_w - H_s$) varies over the length of the drier and a mean value $(H_w - H_s)_m$ must be used in calculating k_G .

As the stock tends to approach the wet-bulb temperature, the humidity difference at any point can be calculated from the wet- and dry-bulb temperature readings by means of the equation³:

$$H_w - H_s = \frac{0.26}{\lambda_w} \times (t_s - t_w) \quad (2)$$

where

t_s = dry bulb temperature, °F.

t_w = wet bulb temperature, °F.

λ_w = latent heat of vaporization of water at t_w , B.t.u./lb.

For example, at Station No. 1:

$t_s = 81.3^\circ\text{F.}$

$t_w = 71.7^\circ\text{F.}$

$H_w = 0.0170 \text{ lb. water/lb. dry air at } 71.7^\circ\text{F. (from humidity chart).}$

$\lambda_w = 1051 \text{ B.t.u./lb. at } 71.7^\circ\text{F. (from steam table).}$

Therefore,

$$H_w - H_s = \frac{0.26}{1051} \times (81.3 - 71.7) = 0.0024 \text{ lb. water/lb. dry air.}$$

In order to calculate the mean humidity difference, the values of the humidity differences at the various stations were calculated and plotted in Figure 3 as a function of the distance from the air exit. This curve was graphically integrated, and the value of the integral was divided by the distance between the first and last thermometer stations—18.435 ft.; this gave a value of 0.0064 for $(H_w - H_s)_m$.

Substituting the values of $\frac{dw}{d\theta}$, $(H_w - H_s)_m$, and A in Equation (1):

$$k_{a'} = 1.023 \times \frac{1}{0.0064 \times 9.36}$$

$$= 17.1 \text{ lb. water evaporated per hr. per ft.}^2 \text{ per unit humidity difference.}$$

Calculation of the Mass Velocity of the Air. This is calculated from the difference in the humidity between the entering and exit air, the quantity of water evaporated per hour, and the cross-sectional area of the tunnel with the pans in place.

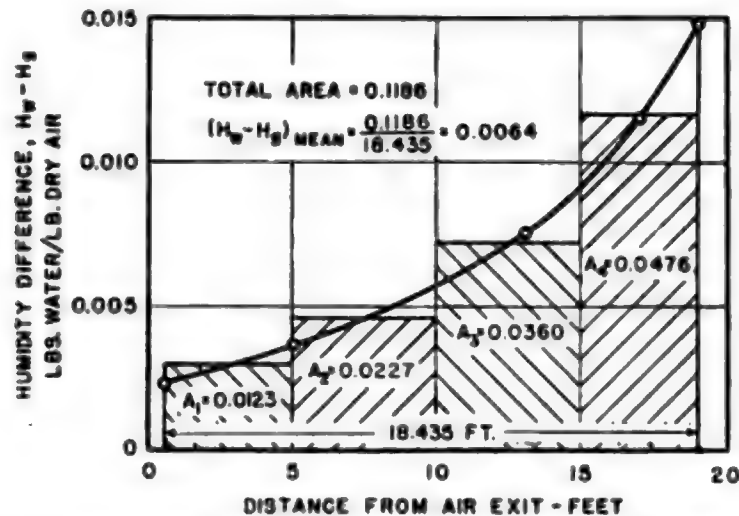


Figure 3. Calculation of Mean Humidity Difference in Tunnel Drier

1. *Humidity of Entering Air, H_1 .* This is calculated from Equation (2), as follows

$$H_1 = H_w - \frac{0.26}{\lambda_w}(t_g - t_w)$$

$$= 0.0215 - \frac{0.26}{1048}(138.9 - 79.0)$$

$$= 0.0066 \text{ lb. water/lb. dry air.}$$

where

t_g = dry-bulb temperature of the air at Station No. 6 = 138.9°F.
 t_w = wet-bulb temperature of the air at Station No. 6 = 79.0°F.
 λ_w = latent heat of vaporization of water at 79.0°F. = 1048 B.t.u./lb.
 H_w = saturation humidity of air at 79.0°F. = 0.0215 lb. water/lb. dry air

2. *Humidity of Exit Air, H_2 .* This is also calculated from Equation (2):

$$H_2 = 0.0170 - \frac{0.26}{1051}(81.3 - 71.7)$$

$$= 0.0146 \text{ lb. water/lb. dry air.}$$

where

81.3 = t_g = dry-bulb temperature of air at Station No. 1, °F.

71.7 = t_w = wet-bulb temperature of air at Station No. 1, °F.

0.0170 = H_w = saturation humidity of exit air at 71.7°F., lb. water/
lb. dry air.

1051 = λ_w = latent heat of vaporization of water at 71.7°F.
B.t.u./lb.

Therefore,

$$\begin{aligned} G &= \frac{dw}{d\theta} \times \frac{1}{(H_2 - H_1)} \times \frac{1}{\text{cross-sectional area}} \\ &= 1.023 \times \frac{1}{(0.0146 - 0.0066)} \times \frac{1}{0.108} \\ &= 1,185 \text{ lb. dry air}/(\text{hr.} \times \text{ft.}^2). \end{aligned}$$

NOMENCLATURE

A = area of drying surface, ft.²

$\frac{dw}{d\theta}$ = drying rate, lb./hr.

G = mass velocity, lb. dry air/(hr. \times ft.²).

H_1 = humidity of entering air, lb. water/lb. dry air.

H_2 = humidity of exit air, lb. water/lb. dry air.

H_g = humidity of the air, lb. water/lb. dry air.

H_w = saturation humidity at wet bulb temp., lb. water/lb. dry air.

$k_{G'}$ = drying coefficient, lb. water evaporated per hr. per ft.² per unit
humidity difference.

t_g = dry bulb temp. °F.

t_w = wet bulb temp. °F.

λ_w = latent heat of vaporization of water at t_w , B.t.u./lb.

Subscript

m = mean.

REFERENCES

1. BADGER AND McCABE, "Elements of Chemical Engineering," McGraw-Hill Book Company, New York, 2nd ed., 1936, p. 308.
2. Ibid., pp. 312-8.
3. Ibid., pp. 253-4.

A STEAM-HEATED AGITATED CONTINUOUS DRIER

Designed and Constructed by

The Department of Chemical Engineering
University of Texas*

Description. This drier consists of an inclined, stationary, steam-jacketed cylinder provided with a rotating, paddle-type stirrer which is driven by an electric motor through a gear reducer, a chain, and sprockets. Feed is introduced continuously at one end of the drier through a hopper and a chain-driven, steam-jacketed screw feeder which is rotated by means of a separate motor. The detention period can be varied by raising or lowering the discharge end of the drier. The construction of the drier is shown in Figures 1 and 2.

Materials and Cost. The cost of the materials used in constructing this drier, as listed in Table 1, amounted to about \$175, and the cost of labor for fabrication and erection was \$223.

Notes on Construction. 1. The body of the drier is a 9-foot section of 16-inch O.D. *A.P.I.* line pipe enclosed in a rolled and welded cylinder of 12-gage sheet steel, 18-inches I.D. and $8\frac{1}{2}$ feet long. Three $\frac{1}{2}$ x 7-inch spacer bars, $8\frac{1}{2}$ feet long, are welded to the steel pipe, as shown in Figure 2, to hold the outer shell in place. To permit proper steam circulation, $\frac{1}{2}$ -inch holes are drilled in the spacer bars every 3 inches along their lengths.

2. The ends of the annular space are closed by welding on rings of 12-gage steel, 16-inch I.D. x $18\frac{1}{2}$ -inch O.D. Flanges of $\frac{3}{4}$ -inch steel, 16-inch I.D. x 20-inch O.D., that have been drilled and tapped to take five $\frac{1}{2}$ -inch bolts, are welded to each end of the inner pipe to hold the cover plates. These plates are steel disks, 20-inch O.D. x $\frac{3}{4}$ -inch thick; they are bolted in place with rubber gaskets to make them tight.

3. The shaft is a $9\frac{1}{4}$ -foot length of extra strong 2-inch steel pipe, which rotates in ball bearing assemblies attached to the centers of the cover plates. Four pairs of hinged blades, constructed as shown in Detail A of Figure 2, are mounted on the shaft. Adjacent stirrer units are mounted with their axes through the spokes at right angles to each other.

4. The screw feeder (Detail D) is a 2-foot section of a commercial 6-inch steel screw conveyor with a jacketed steel trough. A hopper, fabricated from 22-gage steel, is mounted above the feeder.

* Reported by W. A. Cunningham.

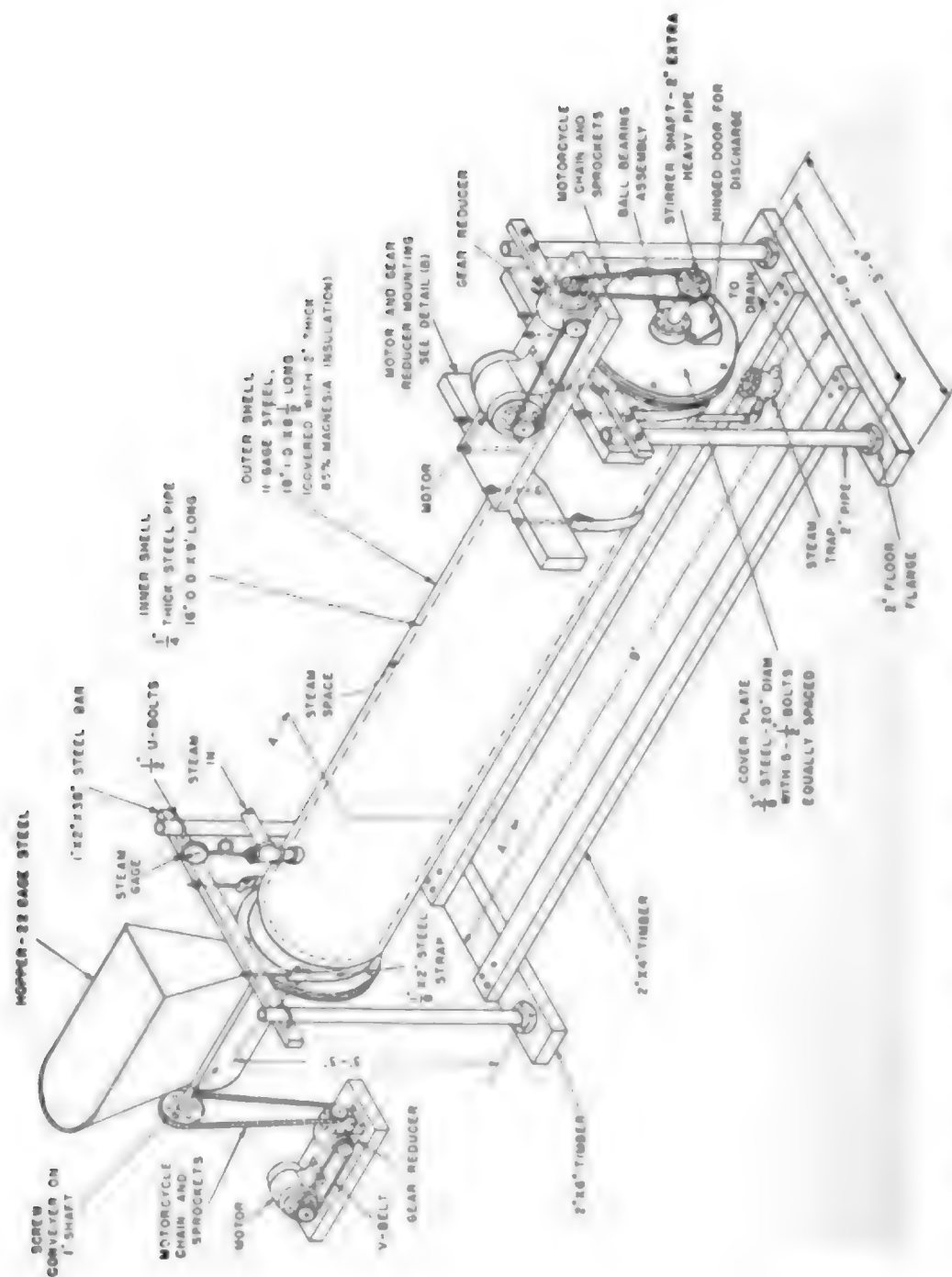


Figure 1. Assembly of Steam Heated Agitated Continuous Drier—University of Texas



5. The drive motor and gear reducer are mounted on top of the drier on a support, constructed as shown in Detail B, and the drier is supported at both ends by the arrangement shown in Figure 1.

5. The drive motor and gear reducer are mounted on top of the drier on a support, constructed as shown in Detail B, and the drier is supported at both ends by the arrangement shown in Figure 1.

TABLE 1
Bill for Materials for Steam-Heated Continuous Drier
 University of Texas

Quantity	Description	Size	Material
16 ft.	Std. pipe	2 in.	Black wrought steel
15 in.	Extra strong pipe	2½ in.	Steel
10 ft.	" " "	2 in.	"
9 ft.	A.P.I. line pipe	16 in. O.D. x 15½ in. I.D.	Black wrought steel
1	Cylinder	18 in. I.D. x 8½ ft. long x 12-gage	Steel
1	Std. nipple	1 in. x 6 in. long	Black wrought steel
1	" "	1 in. x 4 in. "	" " "
1	" "	½ in. x 8 in. "	" " "
2	" "	½ in. x 3 in. "	" " "
1	" "	½ in. x 6 in. "	" " "
1	Std. 90° elbow	½ in.	" " "
1	" " "	½ in.	" " "
1	" 45° "	1 in.	" " "
1	Ground-joint union	½ in.	" " "
2	Disk	20 in. diam. x ½ in. thick	Steel
2	Ring	16 in. I.D. x 20 in. O.D. x ½ in. thick	"
2	"	16 in. I.D. x 18½ in. O.D. x 12-gage	"
1 sq. ft.	Plate	½ in.	"
7 sq. ft.	Sheet	22-gage	"
7½ ft.	Bar	1 x 2 in.	"
12 ft.	Flats	½ x 1 in.	"
25½ ft.	"	½ x ½ in.	"
18 ft.	"	1½ x 2 in.	"
18 ft.	"	½ x 2 in.	"
18 ft.	"	½ x 1½ in.	"
2	Ball-bearing assembly	For 2½ in. shaft	"
20 ft.	Rod	½ in.	Cold rolled steel
4	Floor flange	2 in.	Black malleable iron
2	Bushing	1½ in. I.D. x 3½ in. O.D. x 3 in. long	Brass
1	Std. gate valve	1 in.	"
1	Pressure gage	0 to 100 lb., 3½ in. face	"
1	Steam-gage siphon	½ in.	Black wrought steel
1	Screw conveyor and jacketed trough	6 in. diam. x 2 ft. long	Steel
1	Steam trap	½ in.	Cast iron
10	Bolts	½ in. x 2 in. long	Steel

TABLE 1—*Concluded*

Quantity	Description	Size	Material
12	Bolts & nuts	$\frac{1}{2}$ in. — 13 NC x 8 in. long	Steel
24	" " "	$\frac{1}{2}$ in. — 20 NC x 3 in. long	"
24	" " "	$\frac{1}{2}$ in. — 20 NC x 1 in. long	"
24	Nuts	$\frac{1}{2}$ in. — 13 NC	"
18	Set screws	$\frac{1}{2}$ in. x 1 in. long	"
36	Washers	$\frac{1}{2}$ in.	"
36	"	$\frac{1}{2}$ in.	"
4	Motorcycle sprocket	—	"
2	Motorcycle chain	—	"
2	Gear reducer	—	—
1	Electric motor	1 H.P.	—
1	" "	$\frac{1}{2}$ H.P.	—
2	V-belt	—	Rubber
1	Timber	2 x 12 x $4\frac{1}{2}$ in.	Pine
2	"	2 x 6 x 30 in.	"
1	"	2 x 4 x 38 in.	"
2	"	2 x 6 x 38 in.	"
2	"	2 x 4 x 9 ft.	"
2	Gasket	15 $\frac{1}{2}$ in. I.D. x 20 in. O.D. x $\frac{1}{8}$ in. thick	Rubber
1 bag	Insulating cement	—	85% magnesia

6. The outside of the drier is covered with 2 inches of 85% magnesia insulating cement.

TYPICAL EXPERIMENT

Object. To determine the capacity of the drier and the approximate overall heat-transfer coefficient for the drying of a gypsum plaster from an initial free-moisture content of approximately 30 to 35 per cent, based on the dry weight of $\text{CaSO}_4 \cdot \frac{1}{2}\text{H}_2\text{O}$, down to a final moisture content of less than 1 per cent.

The plaster is a special type of calcium sulfate hemi-hydrate made by heating 30-mesh gypsum in 30 to 35 per cent magnesium sulfate solution, and then filtering and washing the product in a centrifuge. The hot product leaving the centrifuge is immediately passed into the drier.

Procedure. 1. Turn on the steam, start the motors for the stirrer and the feeder, fill the feed hopper, and start feeding the hot, wet plaster to the drier.

2. Adjust the rate of feed and the slope of the drier to obtain maximum production and yet reduce the moisture content of the product to less than 1 per cent.

3. At intervals of time, read the steam gage, measure the temperatures of the feed and product, and collect samples for moisture determination. Determine the moisture contents by drying the samples in an oven at 212°F.

4. Calculate the overall heat-transfer coefficient from the experimental results.

Experimental Data. The following average experimental results were obtained for the drying of the plaster:

1. Weight of the wet feed = 125 lb./hr. = w .
2. Temperature of the entering feed = 190°F. = t_1 .
3. Free-moisture content of the product = less than 1 per cent.
4. Water evaporated = 19 lb./100 lb. feed = W .
5. Steam pressure = 30 lb./in.² gage.
6. Steam temperature = 274°F. (from steam table) = t_2 .
7. Mean temperature of plaster in drier = 215°F. = t_3 .
8. Latent heat of vaporization of water at 215°F. = 970 B.t.u./lb. (from steam table) = λ .
9. Inside area of drier = 35.8 ft.² = A .
10. Heat capacity of wet cake = 0.25 B.t.u./lb. = s .

Calculations.

1. *Heat transferred*

$$\begin{aligned}
 &= w \times s \times (t_3 - t_1) + W \times \lambda \times w/100 \\
 &= 125 \times 0.25 \times (215 - 190) + 19 \times 970 \times 125/100 \\
 &= 23,820 \text{ B.t.u./hr.} = q.
 \end{aligned}$$

2. *Overall heat-transfer coefficient*

$$\begin{aligned}
 &= \frac{q}{A \times (t_2 - t_3)} = \frac{23,820}{35.8 \times (274 - 215)} \\
 &= 11.3 \text{ B.t.u./hr.} \times \text{ft.}^2 \times ^\circ\text{F.}
 \end{aligned}$$

A ROTARY DRIER

Designed and Constructed by

The Department of Chemical Engineering
University of Kansas*

Description. A single-pass, direct gas-fired rotary drier in which the feed, introduced through a hopper provided with a manually-operated feeder, is passed countercurrent to heated air and combustion gases. The drier tube is made of a 9-foot section of 12-inch A.P.I. line pipe; it is driven by a $\frac{1}{4}$ -H.P. motor through a gear reducer, motorcycle chain and sprocket.

Details of the assembly of this drier are shown in Figures 1 and 2.

Materials and Cost. A bill of materials for this drier is given in Table 1. The cost of these materials, when purchased new, is approximately \$125. However, a number of the items used in the construction of the drier were salvaged from scrap. These included the section of 12-inch pipe, the steel drum, the pair of roller skates, and the steel angles and tee bars. This materially reduced the cost.

Uses. At the *University of Kansas*, this drier is used in the junior operations laboratory to study the drying of sand; and in the senior process laboratory it is used in connection with the manufacture of salt from Kansas salt brine.

Operation. A weighed quantity of the feed, which has been sampled for moisture determination, is placed in the feed hopper; the portable instruments are set in place; the burner is lighted; and the drier is set in motion. Feed from the hopper is passed into the drier at a regulated rate, and the dried material is collected and weighed.

TYPICAL EXPERIMENT

Object. To determine the operating characteristics and efficiency of a direct-fired rotary drier, and the cost of drying wet sand.

Procedure. 1. Secure the following equipment from the instrument room:

- a. Dry test gas meter for measuring the volume of burner gas.
- b. Anemometer for measuring the volume of flue gas.

* Reported by T. H. Marshall. The drier was designed and constructed as a senior design project by Leonard Fasholtz and John Shaw.

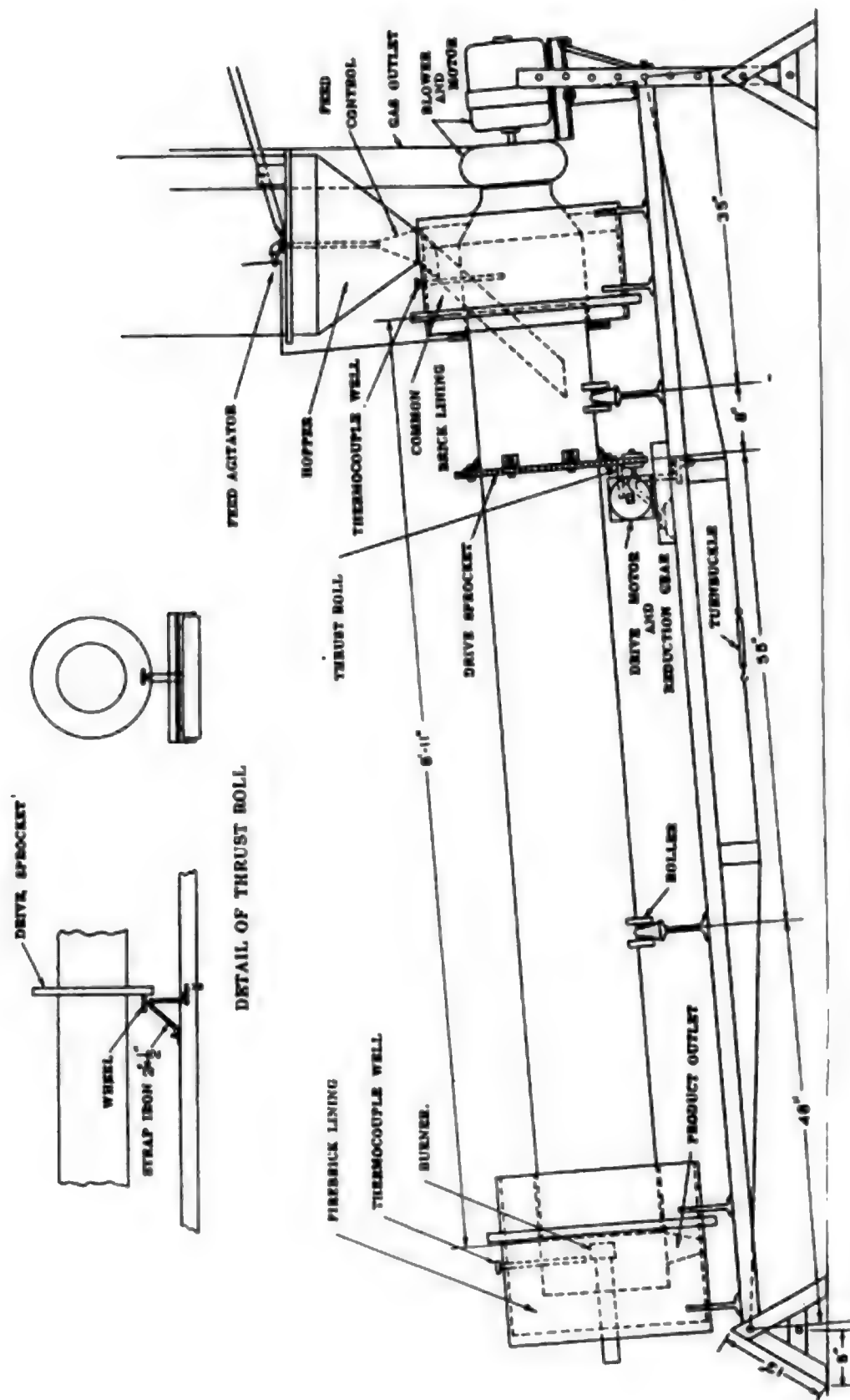


Figure 1. Assembly and Details of Rotary Drier—University of Kansas

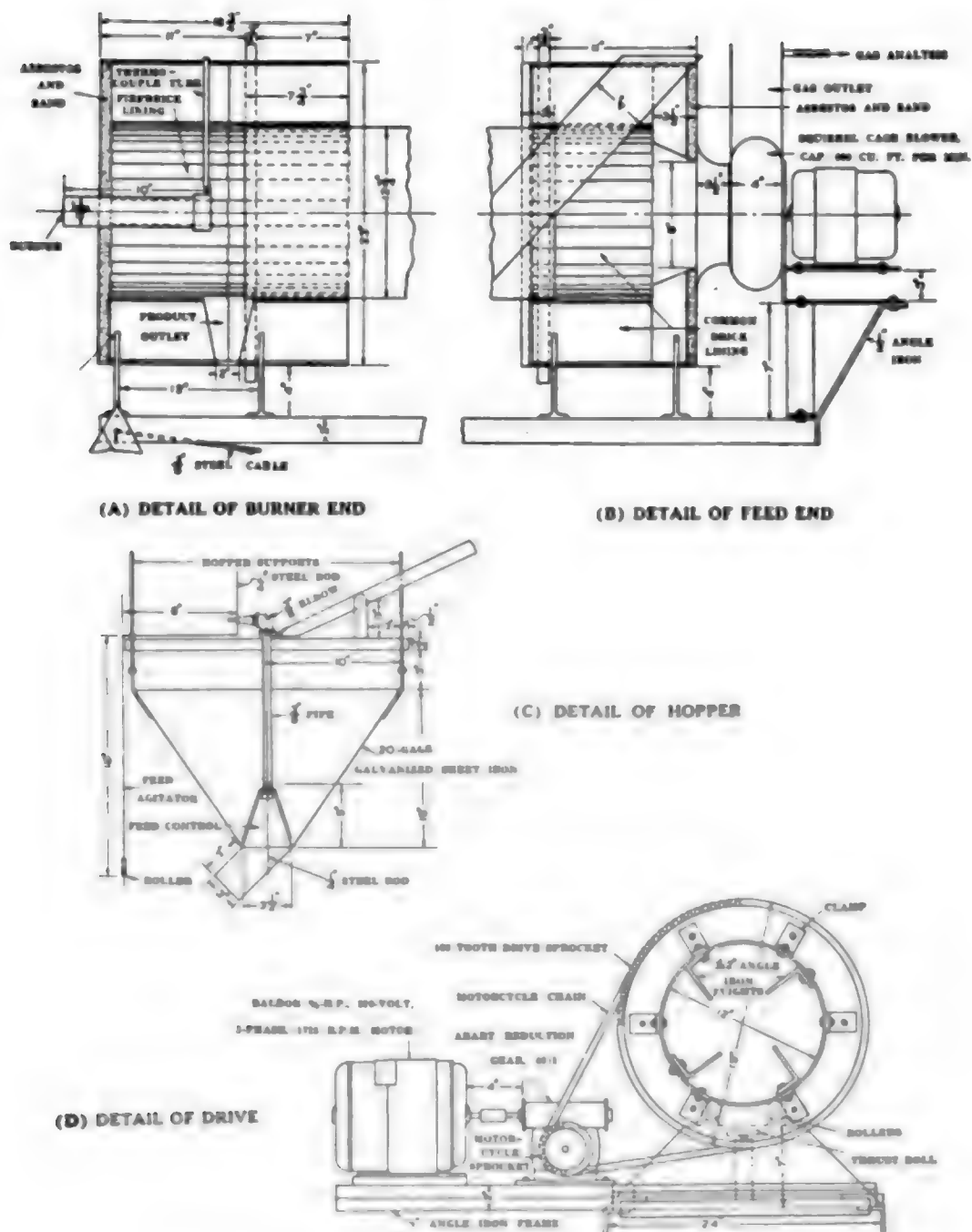


Figure 2. Details of Construction of Rotary Drier

- c. Platform balance and two 5-gallon pails for weighing the feed and the product.
- d. Portable potentiometer for use with the thermocouples.
- e. Orsat apparatus for analyzing the flue gas.
- f. Set of wet- and dry-bulb thermometers for determining the the humidity.

2. Connect the dry gas meter in the line leading to the burner of the drier.

3. Start the drier rotating, turn on the exhausting blower, and light the burner.

TABLE 1
Bill of Materials for Rotary Drier
University of Kansas

Quantity	Description	Size	Material
9 ft.	A.P.I. line pipe	12 in.	Black wrought steel
10 in.	Std. pipe	2 in.	" " "
4 ft.	Flue pipe	4 in. x 20-gage	Galv. steel
1	Std. cap	2 in.	Black malleable iron
36 ft.	Angles	3 x 2 x $\frac{1}{8}$ in.	Steel
45 ft.	"	2 x 2 x $\frac{1}{8}$ in.	"
10 ft.	"	1 x 1 x $\frac{1}{8}$ in.	"
4 ft.	I-beam	8 in.	"
1 piece	Boiler plate (for drive gear)	18 in. O.D. x $\frac{1}{8}$ in. thick	"
1	Drum	55 gal.	"
55	Firebrick	No. 2 arch	—
26	Common brick	Std.	—
1 pair	Roller skates	—	Steel
1	Sheet	4 sq. ft. x 20-gage	Galv. steel
1	Motorcycle sprocket	—	Steel
1	" chain	8 ft.	"
1	Gear reducer	48:1 reduction	"
1	Electric motor	$\frac{1}{2}$ H.P., 1725 R.P.M.	
1	Motor coupling	For $\frac{1}{2}$ in. shaft	Steel
1	Blower and motor assembly	600 ft. ³ /min. capacity	—
3 ft.	Rod	$\frac{1}{2}$ in.	Steel
1 ft.	Flats	2 x $\frac{1}{2}$ in.	"
25 ft.	Thermocouple wire	18-gage	Chromel-alumel
2	Thermocouple protection tube	1 ft. long	Alundum
1	Turnbuckle	16 in. long	Steel
20 ft.	Cable	$\frac{1}{2}$ in.	"

4. Check the operation of the Orsat analyzer, and attach it to the sample thief.

5. Thoroughly mix a predetermined quantity of sand and water, and reserve samples for analytical moisture determinations.

6. Attach the portable potentiometer to the thermocouple lead wires, and examine all switches and lines to insure proper operation. Use the temperature of the room as the cold-junction temperature.

7. Place the weighed, wet sand into the hopper, and feed it through the drier at a rate assigned by the laboratory instructor.

8. After uniform conditions have been established by trial readings of temperatures and weights, make three runs, varying the moisture content of the feed according to instructions from the laboratory instructor.

TABLE 2

Experimental Results of the Drying of Sand in a Rotary Drier

Item	Run No.		
	1	2	3
Wet-bulb temperature of entering drier gas, °F...	80.6	90.0	92.1
Dry-bulb temperature of entering drier gas, °F.....	129.2	170.0	178.3
Wet-bulb temperature of exit drier gas, °F.....	77.0	113.0	117.5
Dry-bulb temperature of exit drier gas, °F.....	98.6	146.0	153.0
Wet-bulb temperature of entering air, °F.....	68.0	65.1	65.7
Dry-bulb temperature of entering air, °F.....	80.6	80.6	74.7
Temperature of wet sand, °F.....	76.0	75.2	75.2
Temperature of dry sand, °F.....	176.0	419.0	471.2
Quantity of wet sand, lb./hr.....	145.4	260.5	148.5
Quantity of dry sand, lb./hr.....	143.0	244.6	133.0
Quantity of drier gas, ft. ³ /hr.....	14,580	5,400	5,070
Quantity of natural gas burned, ft. ³ /hr.....	39.0	58.2	60.0
Heating value of natural gas, B.t.u./ft. ³	946	946	946
Moisture content of wet sand, per cent.....	1.55	6.00	10.45
Moisture content of dry sand, per cent.....	0.00	0.00	0.00
Carbon dioxide in flue gas, per cent.....	—	1.00	1.40
Oxygen in flue gas, per cent.....	20.00	19.00	18.70

* The drier gas consists of the combustion products of the natural gas together with air that enters the end of the drier.

9. Record the following data for each run:

a. Direct or observed data:

- (1) Wet- and dry-bulb temperatures of the entering flue gas.
- (2) Wet- and dry-bulb temperatures of the exit flue gas.
- (3) Wet- and dry-bulb temperatures of the intake gas.
- (4) Temperature of the feed and the product.
- (5) Pounds per hour of feed and product.
- (6) Cubic feet of flue gas per hour.
- (7) Cubic feet of natural gas burned per hour.
- (8) Per cent moisture in feed and product.
- (9) Analyses of flue gas.

b. Chart and other data:

- (1) Humidity and dew points of the dry and wet flue gas.

- (2) Humid volume of the wet flue gas.
 - (3) Average specific heat of the flue gas.
 - (4) Specific heat of dry sand.
 - (5) Cost of natural gas per 1000 cubic feet.
 - (6) Cost of electricity per kw.-hr.
10. Calculate and tabulate the following:
- a. Heat supplied by the burning gas.
 - b. Heat absorbed by the dry sand.
 - c. Heat in the exit flue gas.
 - d. Heat absorbed by the evaporated moisture.
 - e. Thermal efficiency.

TABLE 3
Data from Humidity Chart¹ and Other Sources

Item	Run No.		
	1	2	3
Humidity of entering drier gas, lb. water/lb. dry gas.....	0.0118	0.013	0.014
Dew point of entering drier gas, °F.....	62.0	64.0	67.0
Humidity of exit drier gas, lb. water/lb. dry gas....	0.0130	0.056	0.064
Dew point of exit drier gas, °F.....	64.0	108.0	113.0
Humid volume of exit drier gas, ft. ³ /lb.....	14.2	16.5	17.0
Average heat capacity of drier gas, B.t.u./(lb. × °F.).....	0.244	0.264	0.266
Heat capacity of dry sand, B.t.u./(lb. × °F.).....	0.21	0.21	0.21
Cost of natural gas, dollars/1000 ft. ³	0.28	0.28	0.28
Cost of electricity, dollars/kw.-hr.....	0.065	0.065	0.065
Horsepower required for drier.....	1.5	1.5	1.5

- f. Drying efficiency.
- g. Pounds of water evaporated per pound of fuel.
- h. Pounds of water evaporated per hour.
- i. Capacity of the drier in pounds of sand dried per hour.
- j. Cost of drying the sand.

Typical Data. The experimental results for three runs are tabulated in Table 2; data obtained from the humidity chart and from other sources are given in Table 3; and a summary of the calculated results is given in Table 4.

Calculations for Run No. 3.

1. Heat Supplied by the Burning Gas.

Quantity of natural gas burned, ft.³/hr. = 60.0

Heating value of the gas, B.t.u./ft.³ = 946

$60 \times 946 = 56,760$ B.t.u./hr.

2. Heat Absorbed by the Dry Sand.

Quantity of dry sand, lb./hr. = 133.0

Heat capacity, B.t.u./(lb. \times °F.) = 0.21

Change of temperature, °F. = 471.2 - 75.2 = 396.0

$133.0 \times 0.21 \times 396.0 = 11,100$ B.t.u./hr.

3. Heat Absorbed by the Drier Gas.

Volume of drier gas, ft.³/hr. = 5,070

Humid volume, ft.³/lb. = 17.0

Heat capacity, B.t.u./(lb. \times °F.) = 0.266

Change of temperature, °F. = 153.0 - 74.7 = 78.3

$(5,070/17.0) \times 0.266 \times 78.3 = 6,175$ B.t.u./hr.

TABLE 4

Summary of Results of the Drying of Sand in a Rotary Drier

	Run No.		
	1	2	3
Heat supplied by the burning gas, B.t.u./hr.....	36,900	55,057	56,760
Heat absorbed by the dry sand, B.t.u./hr.....	3,003	17,650	11,100
Heat absorbed by the drier gas, B.t.u./hr.....	7,520	5,690	6,175
Water evaporated, lb./hr.....	2.4	15.9	15.5
Heat absorbed by the evaporated moisture, B.t.u./hr.....	2,425	17,970	16,800
Heat accounted for, B.t.u./hr.....	12,948	41,310	34,075
Thermal efficiency, per cent.....	36.1	75.0	60.0
Drying efficiency, per cent.....	18.7	43.5	49.5
Pounds water evaporated per pound of fuel....	1.23	5.46	5.17
Drying cost, dollars/100 lb. sand (dry basis)....	0.059	0.036	0.066

4. Heat Absorbed by the Evaporated Moisture.

Water evaporated, lb./hr. = 148.5 - 133.0 = 15.5

Enthalpy of water at 75.2°F., B.t.u./lb. = 44.42

Enthalpy of water vapor at 153.0°F., B.t.u./lb. = 1,126.8

$15.5 \times (1,126.8 - 44.42) = 16,800$ B.t.u./hr.

5. Heat Accounted for.

= heat absorbed by sand + drier gas + evaporated moisture

= 11,100 + 6,175 + 16,800 = 34,075 B.t.u./hr.

6. Thermal Efficiency.

$$= \frac{(\text{B.t.u. accounted for})}{(\text{B.t.u. supplied by gas})} \times 100$$

= $(34,075/56,760) \times 100 = 60.0$ per cent.

7. *Drying Efficiency.*

$$= \frac{(\text{B.t.u. absorbed by evaporated moisture})}{(\text{B.t.u. accounted for})} \times 100$$

$$= (16,800/34,075) \times 100 = 49.5 \text{ per cent.}$$

8. *Pounds of Water Evaporated per Pound of Fuel.*

$$\text{Water evaporated, lb./hr.} = 15.5$$

$$\text{Natural gas burned, ft.}^3\text{/hr.} = 60.0$$

$$\text{Density of natural gas, lb./ft.}^3 = 0.05$$

$$15.5/(60.0 \times 0.05) = 5.17 \text{ lb. water/lb. fuel.}$$

9. *Cost of Drying.*

$$\text{Quantity of dry sand, lb./hr.} = 133.0$$

$$\text{Natural gas burned, ft.}^3\text{/hr.} = 60.0$$

$$\text{Cost of natural gas, dollars/1000 ft.}^3 = 0.28$$

$$\text{Power required, H.P.} = 1.5$$

$$\text{Cost of electricity, dollars/kw.-hr.} = 0.065$$

$$\text{Kw.-hr./H.P.} = 0.7457$$

$$100/133.0 \times [(60.0 \times 0.28/1000) + (1.5 \times 0.7457 \times 0.065)]$$

$$= \$0.066/100 \text{ lb. sand (dry basis).}$$

10. *Check Calculations.*a. *Water evaporated, calculated from humidity change.*

$$\text{Volume of drier gas, ft.}^3\text{/hr.} = 5,070$$

$$\text{Humid volume, ft.}^3\text{/lb.} = 17.0$$

$$\text{Humidity change, lb. water/lb. dry gas}$$

$$= 0.064 - 0.014 = 0.050$$

$$(5,070/17.0) \times 0.050 = 14.90 \text{ lb./hr.}$$

b. *Water evaporated, calculated from moisture content of wet sand.*

$$\text{Quantity of wet sand, lb./hr.} = 148.5$$

$$\text{Moisture content of the wet sand, per cent} = 10.45$$

$$\frac{148.5 \times 10.45}{100} = 15.5 \text{ lb./hr.}$$

REFERENCE

1. BADGER AND McCABE, "Elements of Chemical Engineering," McGraw-Hill Book Co., 2nd ed., 1936.

SECTION 5

DISTILLATION

INTRODUCTION

The separation of a liquid mixture into all or a part of its constituents by heating the mixture and recovering the vapors by condensation is known as *distillation*. The process depends upon differences in the vapor pressures of the individual constituents: as the mixture is heated, vapors are evolved which are relatively richer in the more volatile of the constituents, i.e., those of lower boiling points, and, therefore, the condensation of these vapors will give a product having a different composition than that of the original mixture. Obviously, the greater the difference in the vapor pressures of the individual constituents, the greater will be the difference in their concentrations in the vapor phase, and the easier will be the separation. However, the degree of separation depends not only upon the properties of the constituents but also upon the method employed in distilling the mixture. Furthermore, as the number of constituents of the mixture increases, the more difficult it becomes to effect their separation. Therefore, most laboratory distillation experiments are performed with a binary mixture.

Raoult's and Henry's Laws. When two mutually soluble liquids are mixed, each component lowers the vapor pressure of the other. For practically ideal solutions at all concentrations, and for many non-ideal solutions at low concentrations (dilute solutions), the extent of the lowering of the vapor pressure can be determined from *Raoult's law*, which states that at any given temperature the partial pressure of any constituent in a mutually soluble liquid mixture is equal to the product of its vapor pressure in the pure state and its mol fraction in the solution. Mathematically, this relationship may be expressed as

$$p_A = \frac{N_A}{N_A + N_B} P_A = x_A P_A \quad (5.1)$$

and

$$p_B = \frac{N_B}{N_A + N_B} P_B = (1 - x_A)P_B = x_B P_B \quad (5.2)$$

where

p_A and p_B = the partial vapor pressure of components A and B, respectively.

P_A and P_B = the vapor pressure of components A and B , respectively, in the pure state at the temperature of the mixture.

N_A and N_B = the mols of components A and B , respectively, in the mixture.

x_A = mol fraction of component A in the liquid.

x_B and $(1 - x_A)$ = mol fraction of component B in the liquid.

Raoult's law holds only for a mixture that is composed of liquids that do not undergo chemical reaction or molecular association when they are mixed, and whose molecules are approximately the same size. Some mixtures, such as toluene and benzene, and carbon tetrachloride and toluene, obey this law very closely over the entire range of composition, but acetic acid and water, ethyl alcohol and water, and many other common binary mixtures exhibit marked deviations.

According to *Henry's law*, the partial pressure of the component that is present in the smallest concentration—the solute—is proportional to its mol fraction in the solution, or

$$p_s = Kx_s \quad (5.3)$$

where

p_s = the partial vapor pressure of the solute.

x_s = the mol fraction of the solute.

K = a constant that varies with the temperature of the solution, and which must be determined experimentally for each solution.

An inspection of Equations (5.1) and (5.3) shows that Raoult's and Henry's laws differ only by the proportionality constant, and may, therefore, be considered to be special cases of each other; for ideal solutions, the two laws become identical.

Vapor Pressures of Completely Miscible Mixtures. Homogeneous binary mixtures may be classified into three groups:

1. Mixtures with vapor pressures intermediate between those of the pure constituents throughout the entire range of composition.
2. Mixtures with vapor pressures over a certain range of composition greater than that of either of the pure constituents.
3. Mixtures with vapor pressures over a certain range of composition lower than that of either of the pure constituents.

The equilibrium relationships between the liquid and vapor phases of such mixtures can be represented graphically in several different ways, but the most useful of these are the boiling-point and the equilibrium diagrams.

Boiling Point-Composition Diagrams. In this type of diagram, the

boiling points over the entire range of composition of the mixture from that of pure component *A* to that of pure component *B* are plotted as ordinates, and the corresponding liquid compositions as abscissas. On the same graph, a second curve is plotted, which gives the composition of the vapor that is in equilibrium with a given liquid composition at the corresponding temperature. Typical boiling-point diagrams for binary mixtures of the three types are shown in Figure 1.

The upper diagram of Figure 1 is for the system benzene-toluene, and is representative of binary mixtures that obey Raoult's law over the entire range of composition. As the lower curve represents the composition of the liquid and the upper curve represents the equilibrium composition of the vapor, a mixture consisting of 40 mol per cent benzene and 60 mol per cent toluene, for example, boils at a temperature of 203°F.—point *A'*—and the composition of the vapor in equilibrium with this mixture is represented by point *B'*; if this vapor were removed and condensed, the resulting liquid would have a composition of 62 mol per cent benzene and 38 mol per cent toluene—point *C'*.

The data necessary to plot a boiling point diagram for a mixture of this type can be calculated by means of Raoult's law from the vapor pressures of the constituents in the pure state if the mixture can be considered to be ideal. Since the total pressure *P'* of a mixture at any given temperature is equal to the sum of the partial pressures of its constituents, Equations (5.1) and (5.2) may be combined:

$$P' = p_A + p_B = P_A x_A + P_B (1 - x_A) \quad (5.4)$$

which, at a pressure of one atmosphere, becomes*

$$1 = P_A x_A + P_B (1 - x_A) \quad (5.5)$$

or

$$x_A = \frac{1 - P_B}{P_A - P_B} \quad (5.6)$$

Furthermore, according to *Dalton's law of partial pressures*, the mol fraction *y_A* of component *A* in the vapor is

$$y_A = \frac{p_A}{p_A + p_B} \quad (5.7)$$

$$= \frac{P_A x_A}{P'} \quad (5.8)$$

which, at a pressure of one atmosphere, becomes

$$y_A = P_A x_A \quad (5.9)$$

* In these equations, all pressures are in atmospheres.

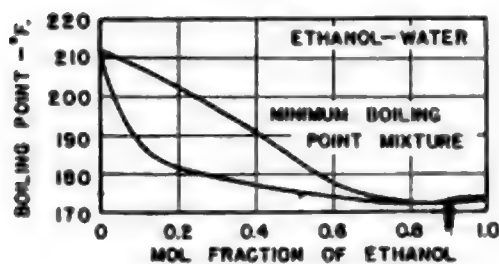
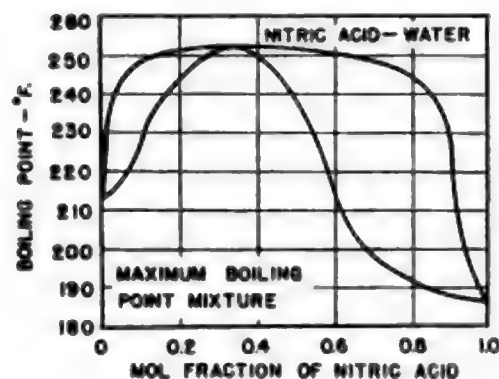
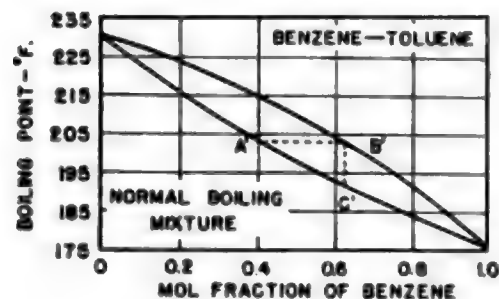


Fig. 1

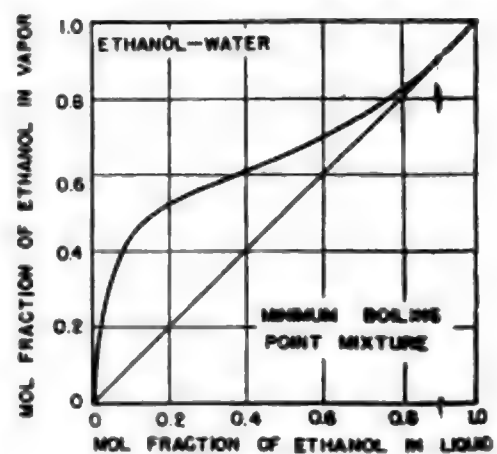
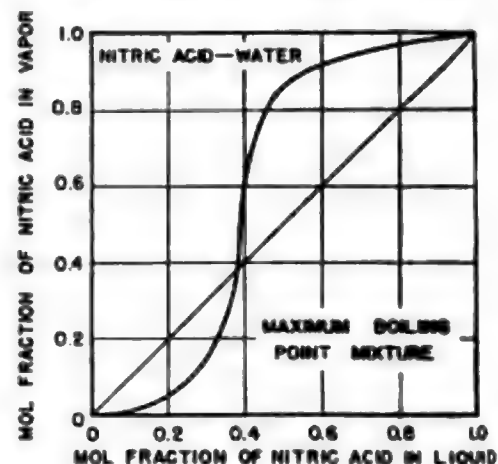
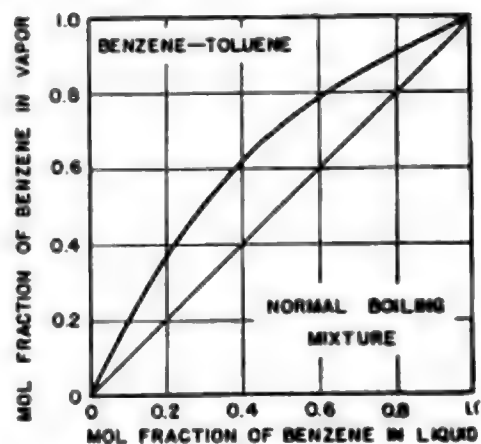


Fig. 2

Figure 1. Boiling Point-Composition Diagrams for Three Types of Completely Miscible Binary Mixtures at a Pressure of One Atmosphere

Figure 2. Equilibrium Diagrams for Three Types of Completely Miscible Binary Mixtures at a Pressure of One Atmosphere

The middle and lower diagrams of Figure 1 are representative of binary mixtures that deviate from Raoult's law: mixtures of the nitric acid-water type exhibit a maximum boiling point, and those represented

by ethanol-water show a minimum boiling point. When a mixture having a composition corresponding to either a maximum or a minimum boiling point is heated, the vapor evolved has the same composition as the liquid; therefore, no separation of the components by distillation is possible. Such binary mixtures are, therefore, known as constant boiling, or azeotropic, mixtures.

Equilibrium Diagrams. A more direct method of representing the equilibrium relationship between the composition of the liquid and the vapor is shown in Figure 2. In this type of diagram, which is known as an equilibrium or an x - y diagram, the concentration of the more volatile component in the liquid phase is plotted against the equilibrium concentration in the vapor phase. It is customary to also include a 45° reference line; a point on this line, obviously, would represent a condition where the compositions of the vapor and liquid phases were identical.

Equilibrium diagrams are especially useful in the solution of distillation problems at constant pressure, for the vapor-liquid equilibrium relationships are affected only slightly by moderate changes in pressure, whereas a slight pressure change introduces an appreciable variation in the boiling point-composition relationships used in plotting a boiling-point curve.

If the mixture obeys Raoult's law, the vapor-liquid equilibrium relationship can be calculated by means of Equations (5.4) to (5.9), but as most liquids are by no means ideal, it is usually necessary to determine the relationship experimentally. The methods developed by Othmer⁷, and Langdon and Keyes⁴ are satisfactory for this purpose.

OPERATING METHODS

Distillation methods may be classified as:

1. Equilibrium, or flash, distillation.
2. Simple, or differential, distillation.
3. Rectification.

Equilibrium Distillation. In equilibrium, or flash, distillation, the liquid mixture is fractionally vaporized, and the evolved vapors are kept in intimate contact with the remaining liquid to establish equilibrium conditions between the two phases before the vapors are removed and condensed. The most important commercial application of this method is in the petroleum industry. The oil is pumped continuously through a pipe still in which it is heated under pressure, and it is then passed to a flash chamber where the pressure is reduced and the oil separates into a vapor and a liquid phase.

The fraction of a binary mixture that must be vaporized to give a product of a definite composition when the distillation is carried out under equilibrium conditions, or the compositions of the liquid and vapor phases that will result when a given fraction of the charge is vaporized can be calculated by means of the equilibrium relationship and a material balance:

$$F x_f = Vy + (F - V)x \quad (5.10)$$

where

F = total mols of feed.

x_f = mol fraction of the more volatile component in the charge.

V = total mols vaporized.

y = mol fraction of the more volatile component in the vapor.

x = mol fraction of the more volatile component in the residual liquid.

The fraction $\frac{V}{F}$ of the charge that must be vaporized in order to obtain a product of a desired composition y can be determined by reading from the equilibrium curve the value of x corresponding to the given value of y , and then substituting the values of x and y in Equation (5.10) and solving for $\frac{V}{F}$.

To determine the composition of the product y and the composition of the residue x that will result from the vaporization of a given fraction of the charge, Equation (5.10) is plotted on the equilibrium diagram, and the intersection of this curve with the equilibrium curve gives the desired values of x and y .

Simple or Differential Distillation. In this method, the vapors evolved from the heated liquid are removed and condensed as rapidly as they are formed. A distillation carried out in this manner does not result in very effective separation of the constituents unless their boiling points are rather far apart.

The Rayleigh Equation. The degree of separation that can be effected by differential distillation of a binary mixture can be calculated by means of the *Rayleigh equation*, which may be written as

$$2.303 \log_{10} \frac{F}{W} = \int_{x_w}^{x_f} \frac{dx}{y - x} \quad (5.11)$$

where

F = mols of liquid mixture at the beginning of the distillation.

W = mols of liquid at the end of the distillation.

x_f = mol fraction of one of the components in the liquid at the beginning of the distillation.

x_w = mol fraction of the same component in the liquid at the end of the distillation.

x and y = mol fractions of the corresponding component in the liquid and vapor phases, respectively.

The value of the integral in Equation (5.11) can be determined graphically by plotting values of $\frac{1}{y-x}$ against values of x , and measuring the area under the curve between the limits of x_w and x_f . The values of x and y needed to plot this curve are obtained from the equilibrium diagram for the mixture.

Rectification. This is the most important of the distillation methods. It involves the return of all or a part of the condensed vapors to the still in such a manner that the vapors rising from the still come in intimate contact with the downcoming liquid, known as reflux. The intimate contact between the vapor and reflux results in a transfer of heat from the rising vapor to the downcoming reflux, thereby causing the condensation of a part of the less volatile component in the vapor and the vaporization of the more volatile component in the reflux. As a result, the rising vapor becomes enriched with the more volatile component, and separation is materially increased. Rectification may be carried out as a batch process, but continuous operation is much more common.

Partial Condensation. In addition to the separation that is effected when a liquid is partially vaporized, additional separation can be accomplished by only partially condensing the vapors that are evolved. If the vapors are cooled only slightly, that portion which condenses will be relatively richer in the higher boiling component, while the remaining vapor, which can be withdrawn and condensed separately, will be relatively richer in the lower-boiling component.

RECTIFICATION EQUIPMENT

The essential parts of a rectification system are: (1) a still or kettle, (2) a column, and (3) a still head.

The Still. The still should be large enough to hold the desired quantity of charge and yet leave sufficient space to effectively separate the vapors from the liquid. For very small laboratory units, the still is often heated externally by electrical heaters, but for larger equipment the heat is generally supplied by steam through coils in the still. For flexibility of operation, multiple sets of heating coils may be provided,

and to permit the easy removal of the coils in the event that replacement becomes necessary, the construction shown in Figure 3 is well suited for laboratory purposes.

The Column. The column is generally a tall cylinder whose function is to effect intimate contact between the rising vapor and the down-coming stream of reflux; this is accomplished by the use of bubble-cap plates, sieve plates, or a packing material.

Bubble-Cap Columns. A bubble-cap column consists of an upright shell containing a vertical series of horizontal plates, which are generally spaced equally throughout the length of the column. Each plate has one or more vapor pipes, which provide a passage for the rising vapor, and a corresponding number of bubble caps, which fit over the vapor pipes and deflect the vapor beneath the surface of the liquid before it rises to the plate above. One or more overflow pipes are also provided

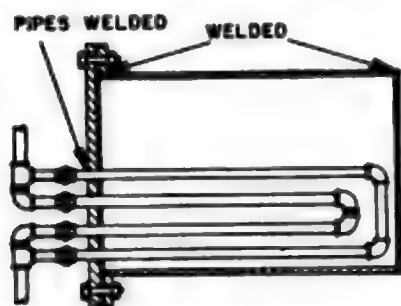


Figure 3. Still with Replaceable Heating Unit

on each plate to maintain a constant liquid level and to provide a passage for the descending reflux. To prevent vapor from rising through the overflow pipe, a liquid seal is made by extending the lower end of the pipe beneath the level of the liquid on the plate below. The same result may also be attained by making a U-bend in the lower end of the pipe.

Several types of bubble caps are available, but the most common cap has a notched or slotted periphery to increase the area of contact and to break up the vapor from the plate below into numerous small bubbles. Bubble caps suitable for laboratory columns may be purchased, or they may be made in the laboratory shops. A simple method for making bubble caps from ordinary 50-cc. iron crucibles is given on page 359. Another type of construction, in which the plate, the bubble cap, the downtake or overflow pipe, and the riser pipe are cast in one piece is shown in Figure 4.*

Certain precautions should be observed in the design of a bubble-cap column:^{1,8,10}

1. The down pipe, or leaving weir, should be located sufficiently far from the last row of caps to provide a quieting area, or some sort of baffling should be employed.
2. The bubble caps should be designed so that the bubbles are not excessive in size.

* Through the courtesy of E. E. Litkenhous.

3. The plates should be designed so that an equal quantity of vapor passes through each of the caps.

4. The overflow should not short-circuit across the plate without properly mixing with the liquid on that plate.

Sieve-Plate Columns. The perforated- or sieve-plate column is constructed with a vertical series of horizontal, perforated plates, each of which is provided with an overflow pipe to maintain a constant depth of liquid on the plate. The descending reflux collects on each plate,

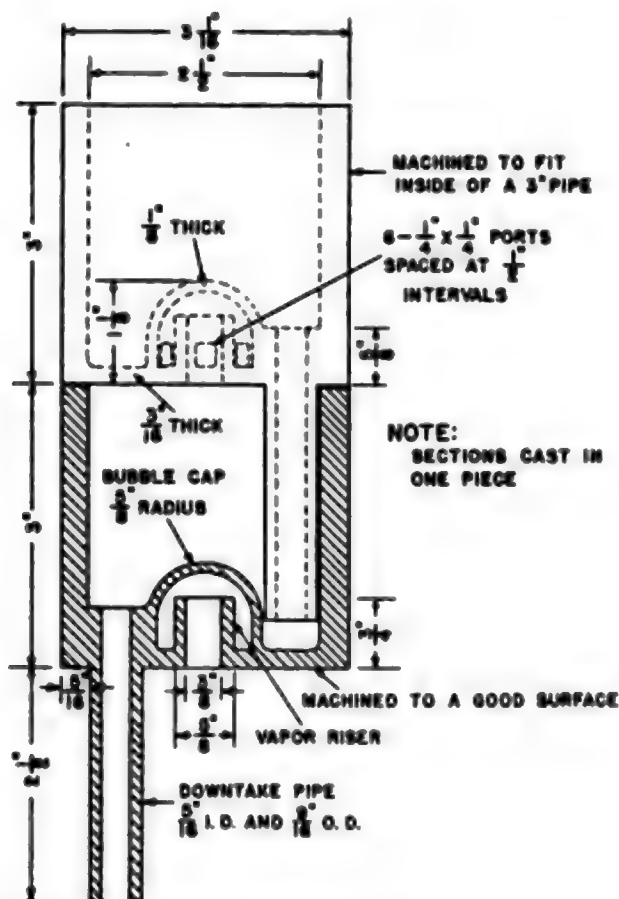


Figure 4. Details of Construction of Bubble-Cap Plate—Vanderbilt University

and the vapors rising through the perforations prevent the liquid from descending to the plate below except through the overflow pipe.

Perforated-plate columns must be operated at a vapor velocity which will be high enough to overcome the static head of the liquid on each plate and thereby prevent the liquid from dropping through the openings, but which will not be so high as to lift the liquid off the plate and thereby decrease the intimate contact between the two streams. Furthermore, the plates must be perfectly level to insure a constant depth of liquid over the entire surface of the plate; if the level is not constant,

the reflux will pass down through the openings in the low side and the vapors will rise through the openings in the high side.

Packed Columns. A column of this type is, essentially, an upright, cylindrical shell filled at random with small pieces of inert solid material. The ascending vapor and the descending liquid come in contact largely on the surface of the packing. Such columns are simple in design, low in cost, and with a suitable packing the effectiveness of the separation per unit height of column compares favorably with that of other types of columns. However, for efficient operation, channeling of the vapors and incomplete wetting of the packing must be avoided. The

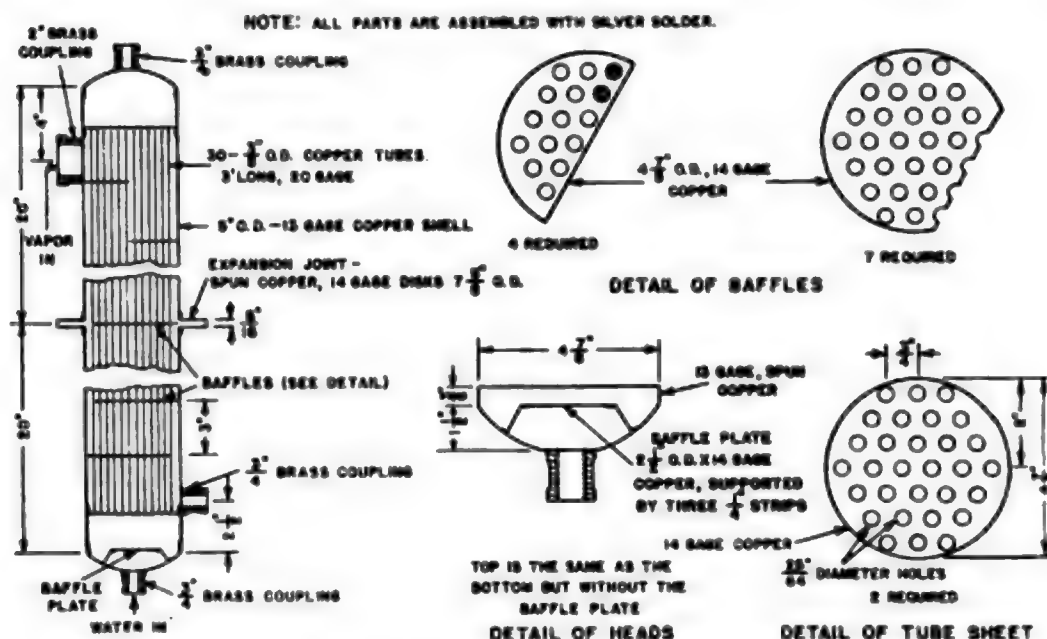


Figure 5. Single-Pass Vertical-Tube Condenser—University of Tennessee

tendency to channel can be reduced¹¹ by making the ratio of the column diameter to the diameter of the individual packing units greater than 8 to 1, and the ratio of the column height to column diameter greater than 15 to 1.

Various types of packing materials may be used in packed columns, including crushed stone, coke, glass beads, metal balls, Raschig rings, Berl saddles, wire gauze, and metal chain. The particular packing selected must be non-reactive with the material being distilled, and it should provide a large contact surface, uniformly distributed interstices, and sufficient free space for a desirable throughput.

The Still Head. That part of the distillation assembly above the column is commonly referred to as the *still head*; this generally includes

a condenser, and some type of device for proportioning the condensate into reflux and product.

The Condenser. The condenser may be designed to effect either partial or total condensation of the vapors from the column. A partial condenser, or dephlegmator, condenses only a portion of the vapors,

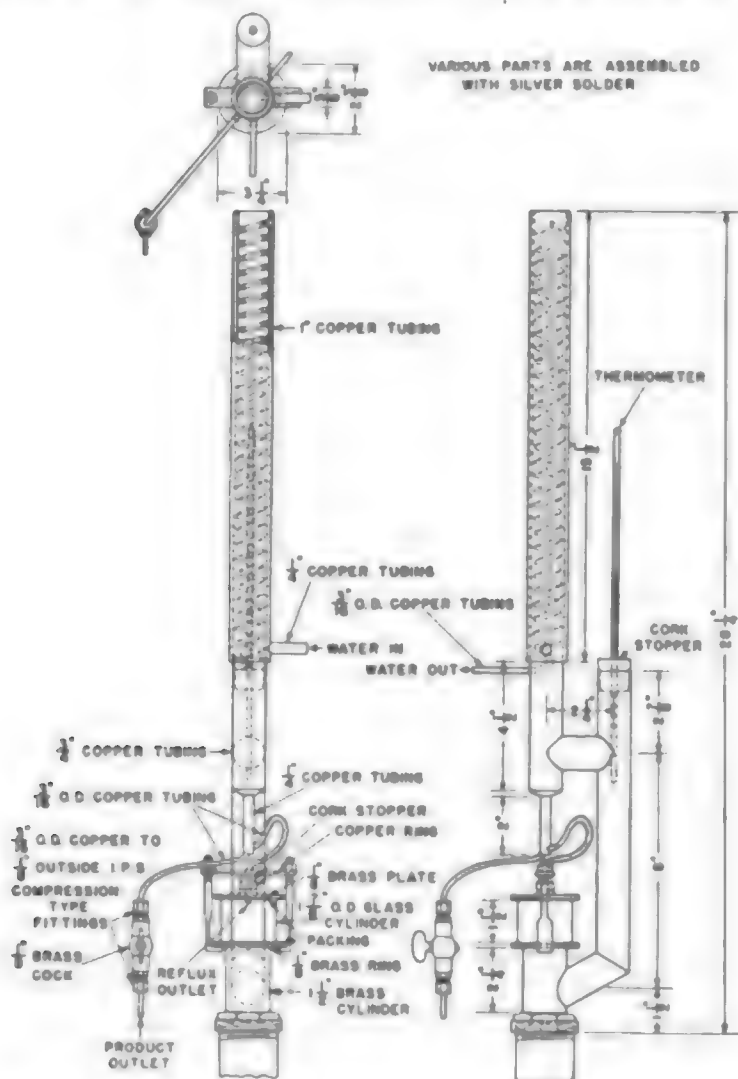


Figure 6. Details of Construction of Column Still Head—University of New Hampshire

which are returned to the column as reflux; the rest of the vapors are delivered to a final, or product, condenser. A total condenser, on the other hand, condenses all of the vapors, and the condensate is returned in total or in part as reflux to the column. In general, a total condenser is preferable because it requires less attention and it simplifies control of the reflux.

A single-pass, vertical-tube condenser suitable for laboratory columns

is shown in Figure 5.* Although such a condenser is characterized by low water velocity and therefore a low overall heat-transfer coefficient, it is simple in design and is easily constructed.

The Proportioning Device. With a total condenser, the reflux ratio, which is the ratio of the quantity of reflux returned to the column to the quantity of product withdrawn, is controlled by a liquid-proportioning device, such as a weir box, or by means of valves and flow meters in the reflux and product lines. In small laboratory columns, difficulty is frequently experienced because of the relatively large hold-up in the proportioning device and the reflux line. For small columns, therefore, a still head, such as shown in Figure 6, is to be preferred. Although the still head shown is for a 2-inch column, the dimensions may be varied to meet individual requirements.

EXPERIMENTAL METHODS

Laboratory rectification experiments are carried out most easily by distilling a binary mixture at total reflux, and from the results of the experiment determining either: (1) the number of theoretical plates in the column, (2) the height equivalent to a theoretical plate (*H.E.T.P.*), (3) the overall plate efficiency, or (4) the *Murphree*, or individual, plate efficiencies. In conducting the experiment, the mixture is distilled until temperature readings indicate that conditions have become stabilized, and then samples are taken for analysis. If only the number of theoretical plates, the *H.E.T.P.*, or the overall plate efficiency is desired, analysis of only the reflux and the still contents is necessary; if individual plate efficiencies are to be determined, samples of liquid and vapor from each plate must also be analysed.

Binary Mixtures for Experimental Studies. Mixtures for laboratory experiments should possess the following properties: (1) the separate constituents should be obtainable in a high state of purity, (2) the mixture should be stable and non-corrosive to the column, (3) the mixture should be capable of easy and accurate analysis, and (4) if the constituents do not form an ideal solution, the vapor-liquid equilibrium data at atmospheric pressure should be available.

Some of the most commonly used binary mixtures are: (1) benzene-toluene, (2) ethanol-water, (3) acetic acid-water, (4) chloroform-toluene, and (5) carbon tetrachloride-benzene.

CALCULATIONS

Number of Theoretical Plates. An ideal column may be considered to be made up of theoretical plates on each of which the vapor and liquid

* Through the courtesy of R. M. Boarts.

are in equilibrium; however, actual columns seldom approach such conditions. One method of evaluating an actual column is to compare the number of actual plates in the column with the number theoretically required to effect the desired separation. The number of theoretical plates for the rectification of a binary mixture is most commonly determined by the McCabe-Thiele graphical method.⁴

McCabe-Thiele Graphical Method. This method is based upon four assumptions, which are generally approximated rather closely in actual practice:

1. *The radiation loss from the column is negligible.* In practice, this is accomplished by proper insulation.

2. *The heat of mixing of the two components is negligible.* This holds true for many mixtures, particularly where the components are chemically similar.

3. *The molal latent heats of the two components are equal.* For mixtures composed of similar types of liquids, this is generally true; the relationship is expressed by *Trouton's rule*, which states that the molal entropy of vaporization of normal liquids is nearly constant. If, however, the molal latent heats are appreciably different, a fictitious molecular weight can be assigned to one of the components so as to make the molal latent heats of the two components equal. In such a case, the data used in plotting the equilibrium diagram must be based on the assumed fictitious molecular weight.

4. *The sensible heat changes throughout the column are small compared to the latent heat.* This is generally true except when there is a large temperature difference between the top and the bottom of the column.

Steps in the McCabe-Thiele Method. This method is illustrated by Figures 7, 8, and 9. The steps are as follows:

1. The equilibrium diagram for the binary mixture is drawn on rectangular coordinate paper (Figure 8).

2. Vertical lines are erected on the diagram corresponding to the compositions of the distillate x_D , the residue x_w , and the feed x_f .

3. The operating lines are drawn. Each operating line has a slope equal to the ratio of mols of reflux to mols of vapor at the particular portion of the column. For operation at total reflux, $V = L$ throughout the column (Diagram A of Figure 7); therefore, the operating line for this case is a 45° diagonal (Diagram A of Figure 8). For continuous rectification at partial reflux, the number of operating lines is one more than the number of feeds; only the conditions for a single feed are illustrated (Figures 7 and 8). In this case, the upper operating line has a

slope of L/V and the lower operating line has a slope of $\frac{L + qF}{V - (1 - q)F}$; these lines intersect on the line $a - b$ (Figure 8). The slope of line

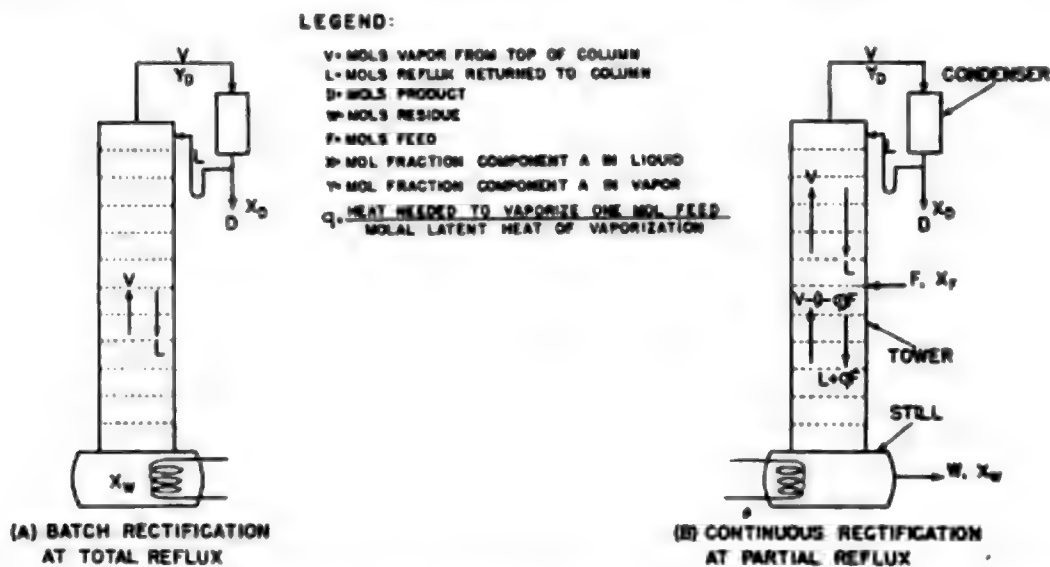


Figure 7. Material Balance for Rectification at Total and at Partial Reflux

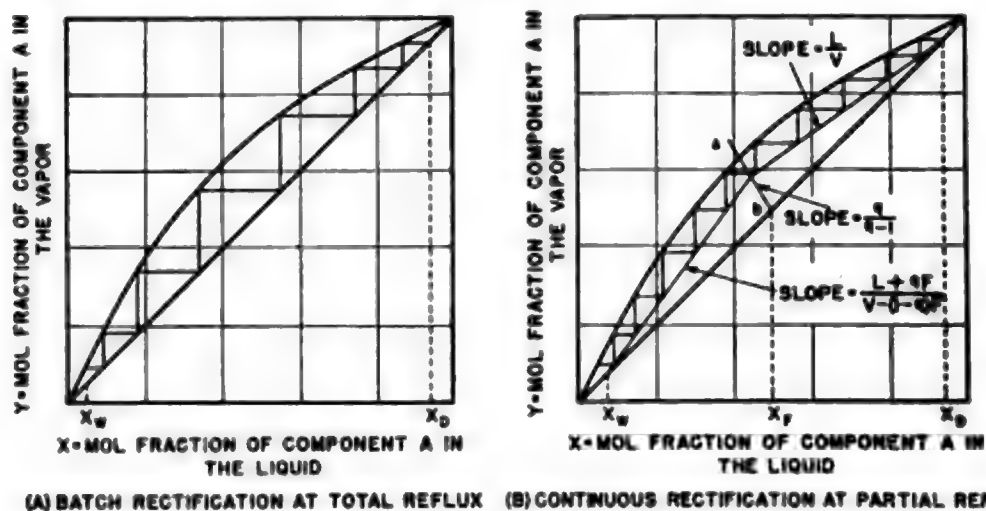


Figure 8. McCabe-Thiele Analysis for Rectification at Total and at Partial Reflux

$a - b$, obviously, depends upon the condition of the feed, for the term q is defined as

$$q = \frac{\text{heat needed to vaporize 1 mol of feed}}{\text{molal latent heat}} \quad (5.12)$$

The effect of the condition of the feed on the slope of line $a - b$, commonly referred to as the q -line, is shown in Figure 9.

4. Starting at the intersection of x_D with the 45° diagonal line, a series of rectangular steps are drawn between the equilibrium curve and the operating lines until x_w is reached. The total number of steps represents the number of theoretical plates in the column and still; the latter is usually considered to be equivalent to one theoretical plate.

Algebraic Methods. When the number of plates in a column is relatively large, that is, in excess of about 30, the graphical method may lead to inaccurate results unless the equilibrium diagram is drawn to a relatively large scale. For such cases, an algebraic method of solution

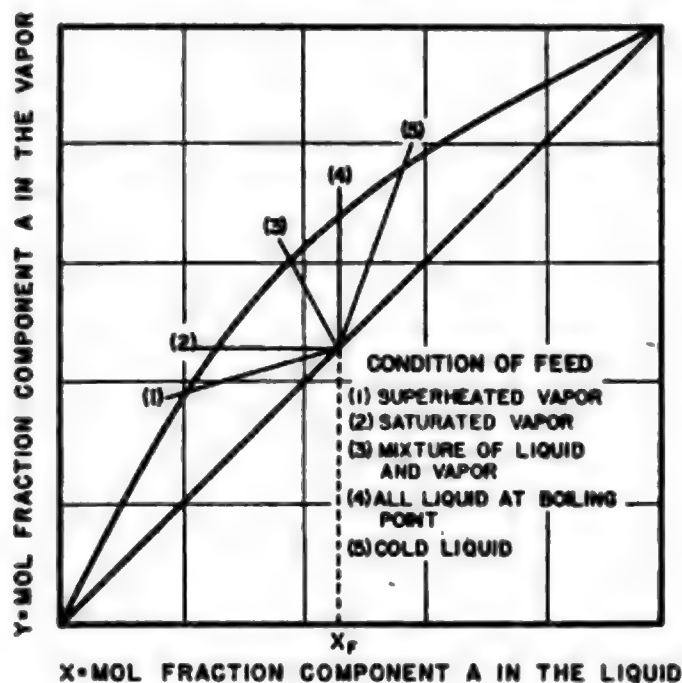


Figure 9. Effect of Feed Conditions on Slope of the q Line

is often more satisfactory. Although a number of algebraic methods have been proposed, the equations developed by Fenske³ and by Dodge and Huffman² for the rectification at total reflux of binary mixtures that obey Raoult's law are probably the most practical. These equations are:

1. *Fenske equation for total reflux*

$$\frac{x_D}{1 - x_D} = (\alpha^{n+1}) \frac{x_w}{1 - x_w} \quad (5.13)$$

which, upon solving for $n + 1$, becomes

$$n + 1 = \frac{1}{\log \alpha} \left[\log \frac{x_D(1 - x_w)}{x_w(1 - x_D)} \right] \quad (5.14)$$

2. *Dodge-Huffman equation for total reflux*

$$n + 1 = \frac{2.303}{\alpha - 1} \left[\log_{10} \frac{x_D}{x_w} + \alpha \log_{10} \frac{1 - x_w}{1 - x_D} \right] \quad (5.15)$$

In these equations

$n + 1$ = number of theoretical plates including the still.

n = number of theoretical plates in the column.

x_D = mol fraction of the more volatile constituent in the distillate.

x_w = mol fraction of the more volatile constituent in the still contents (residue).

α = relative volatility, which is defined by the equation:

$$\alpha = \frac{y_A x_B}{y_B x_A} \quad (5.16)$$

where

x_A and y_A = mol fraction of component A in the liquid and vapor phases, respectively.

x_B and y_B = mol fraction of component B in the liquid and vapor phases, respectively.

In using these equations, the arithmetic average of the relative volatilities of the mixture at the temperature at the top of the column and at the temperature of the still is generally satisfactory, because for most liquid mixtures the ratio of the volatilities does not change greatly with change in temperature. However, for those mixtures where the relative volatility does change appreciably with temperature, the geometric average should be used; this is equal to $\sqrt{\alpha_1 \alpha_2}$, where α_1 is the relative volatility at the temperature at the top of the column and α_2 is the relative volatility at the temperature of the still.⁹

Height Equivalent to a Theoretical Plate (H.E.T.P.). The *H.E.T.P.* is defined as *that height of column which is required to make a separation equivalent to that effected by one theoretical plate.* Mathematically,

$$H.E.T.P. = \frac{\text{Height of column}}{\text{No. of theoretical plates}} \quad (5.17)$$

Overall Plate Efficiency. The overall plate efficiency of a column is *the ratio, in per cent, of the number of theoretical plates divided by the number of actual plates in the column.*

Individual Plate Efficiency. The efficiency of an individual plate is the ratio, in per cent, of the actual change in vapor composition to the change that would occur if complete vapor-liquid equilibrium were reached. This is expressed by the Murphree equation⁶:

$$E_M = \frac{y_n - y_{n+1}}{y_n^* - y_{n+1}} \times 100 \quad (5.18)$$

where

E_M = Murphree plate efficiency.

y_n = mol fraction of the more volatile constituent in the vapor leaving plate n .

y_{n+1} = mol fraction of the more volatile constituent in the vapor rising to plate n .

y_n^* = mol fraction of the more volatile constituent in the vapor leaving plate n if the vapor were in equilibrium with the liquid leaving that plate.

STEAM DISTILLATION

If the mixture being distilled is immiscible with water, or if water constitutes the high-boiling constituent of the mixture, the heat needed for vaporization may be supplied by steam introduced directly into the still instead of into heating coils.

The direct introduction of steam into the charge lowers the partial pressures of the other constituents, and, therefore, causes vaporization to occur at a lower temperature than would be the case if closed-steam heating were employed.

The use of open steam finds its greatest application in a process known as *steam distillation*, in which high boiling materials immiscible with water are separated from nonvolatile impurities at temperatures appreciably lower than would be required for straight distillation.

Since each component of an immiscible liquid mixture exerts the same vapor pressure that it would exert if it were present alone, the mixture will boil at a temperature where the sum of the vapor pressures is equal to the pressure on the system, and the mol fraction of either constituent in the vapor phase will be proportional to its vapor pressure. Therefore, the quantity of water that will distill over with a given quantity of an immiscible liquid B can be calculated from the relationship

$$Wt_A = Wt_B \times \frac{P_A M_A}{P_B M_B} \quad (5.19)$$

where

Wt_A = weight of water in the vapor, lb.

Wt_B = weight of component B in the vapor, lb.

P_A = vapor pressure of water at the temperature of distillation, atm.

P_B = vapor pressure of component B at the temperature of distillation, atm.

M_A and M_B = molecular weight of water and component B , respectively, lb./mol.

NOMENCLATURE

E_M = Murphree plate efficiency.

F = mols of feed.

K = Henry's law constant.

L = mols of reflux.

M = molecular weight, lb./mol.

n = number of theoretical plates.

N = number of mols.

p = partial pressure, atm.

P = vapor pressure, atm.

P' = total pressure, atm.

q = ratio of heat needed to vaporize one mol of feed to molal latent heat.

V = mols of vapor.

W = mols of liquid residue.

Wt = weight, lb.

x = mol fraction of a component in the liquid.

y = mol fraction of a component in the vapor.

y_n^* = mol fraction of more volatile constituent in vapor leaving the n th plate if the vapor were in equilibrium with the liquid leaving that plate.

α = relative volatility.

Subscripts

A = component A .

B = component B .

D = distillate, or product.

f = feed.

n = n th plate in the column.

$n + 1$ = plate immediately below the n th plate in the column.

s = solute.

w = waste, or residue.

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A BUBBLE-CAP RECTIFICATION COLUMN

Designed and Constructed by
The Department of Chemical Engineering
New York University*

Description. This distillation unit consists of a 10-plate bubble-cap column, a steam-heated boiler, a condenser, and the usual accessories. The equipment has been designed so as to provide all possible means for obtaining complete test data, to anticipate all possible variations in operating conditions, and to allow for easy dismantling and reconstruction. Thus: (1) the boiler is a unit in itself, and may be easily detached from the column; (2) the heating surface is composed of three independent units so that the heat input may be varied over a considerable range; (3) the column can be used for batch distillation, or it can be operated continuously with either total or partial reflux; (4) feed can be introduced at any plate, thus permitting even the extremes of operating the column entirely for stripping or for enriching; (5) provision is made for measuring temperatures and rates of flow, and for obtaining samples of liquid and vapor at all vital points in the system; and (6) provision is made for performing distillations under pressure, under vacuum, or with open steam.³

Drawings and Dimensions. Figure 1 is a schematic drawing of the assembled apparatus; details of construction of the plates and the boiler are shown in Figures 2 and 3, respectively; and the most important dimensions of the system are given in Table 1.

Because of the complexity of the piping system, no attempt has been made to show all of the sampling lines. Furthermore, although Figure 1 shows all of the feed lines and sampling tubes in the same vertical plane, this is merely diagrammatic; the position of the various tubes is different on adjacent plates because the down pipes alternate between one side of the column and the other.

Materials and Cost. The materials necessary for the construction of this distillation assembly are listed in Table 2. Their total cost, exclusive of that of the condenser, is estimated to be approximately \$200. This includes \$30 for materials and labor in rolling and welding the boiler, which was done in an industrial shop. The rest of the apparatus was built at odd times over a period of six months by two members of the department and the department mechanic.

* Reported by J. R. Huffman and R. E. Treybal.

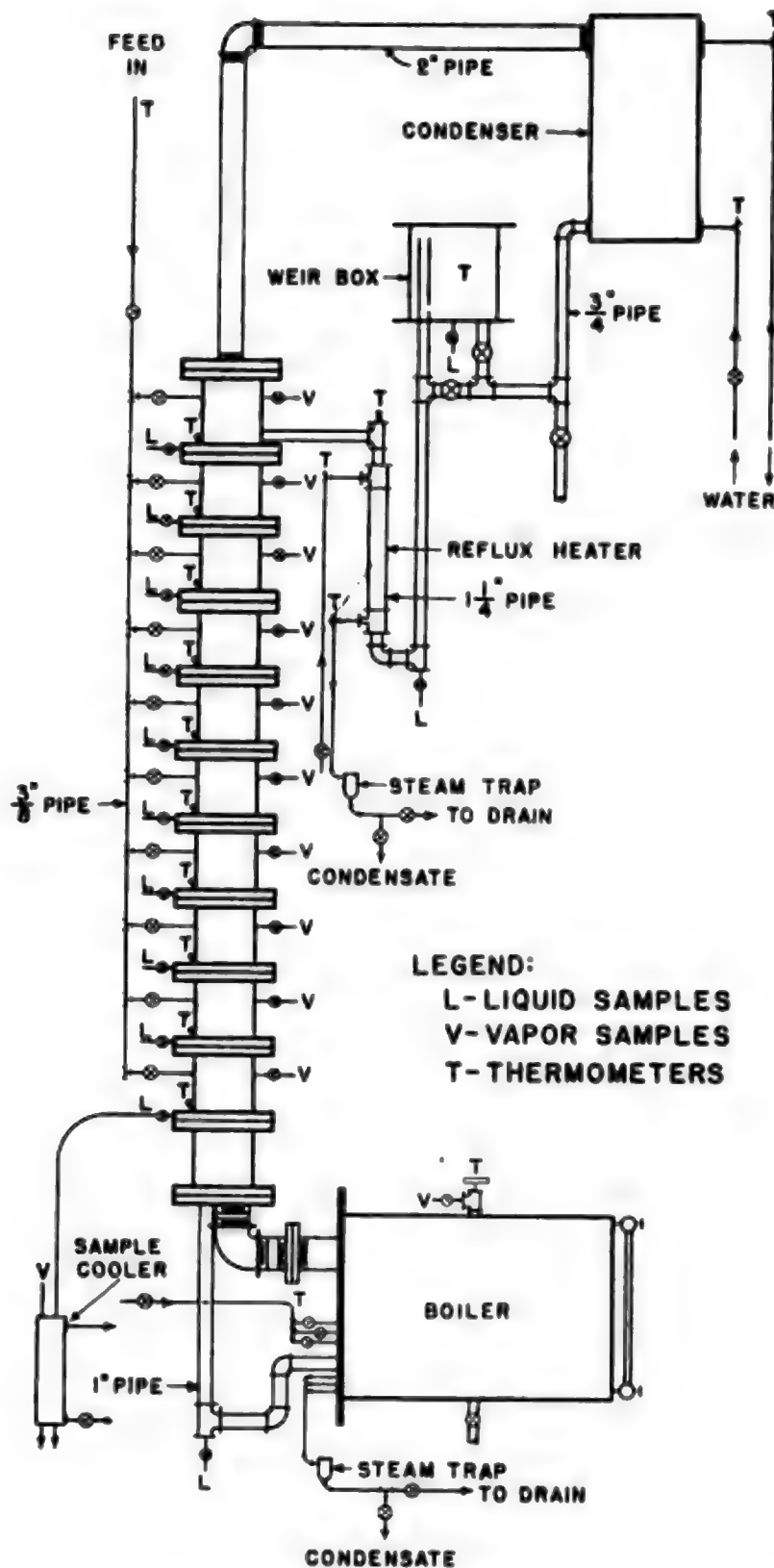


Figure 1. Assembly of Bubble-Cap Rectification Column—New York University

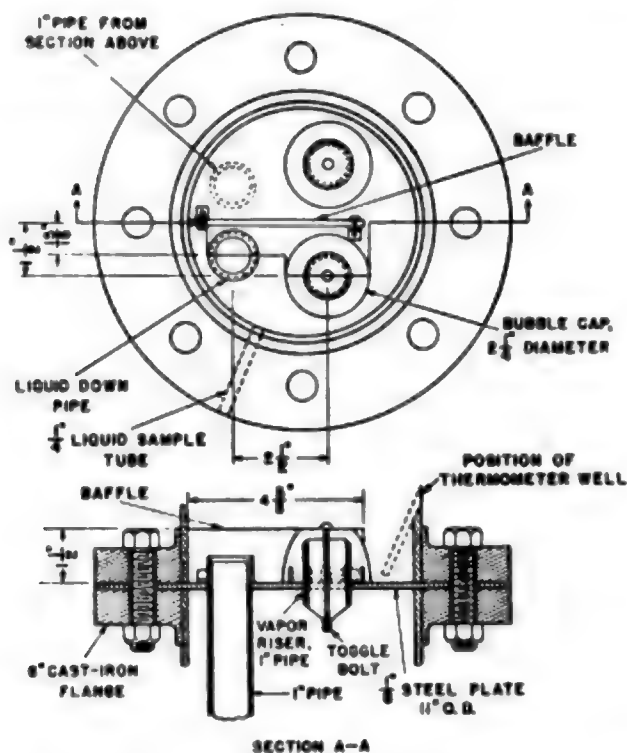


Figure 2. Details of Construction of Bubble-Cap Plate

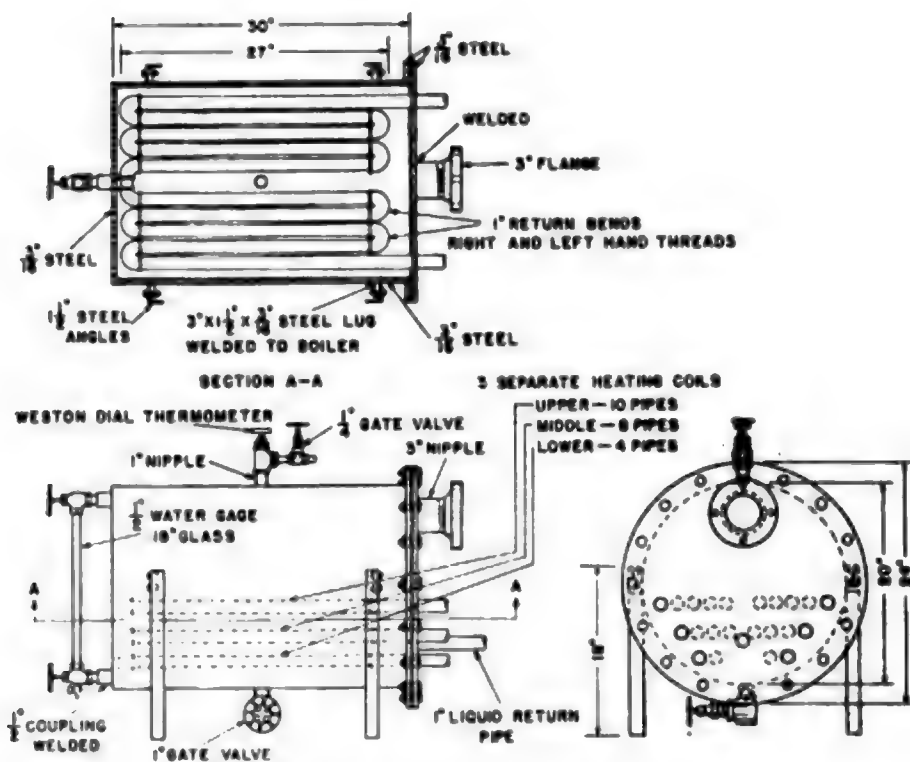


Figure 3. Details of Construction of Boiler

Notes on Construction. The Column. The 10-plate column is built up of separate sections made from standard 6-inch nipples, 8 inches long, and standard 6-inch cast-iron flanges.

The Plates. The plates, which are bolted between the individual sections, are made from $\frac{1}{4}$ -inch cold-rolled sheet steel. Each plate is fitted with two bubble caps made from ordinary 50-c.c. iron crucibles of the type used in analytical work. The slots, twelve to a cap, are $\frac{1}{4}$ -inch high and $\frac{1}{4}$ -inch wide, and are semi-circular at the top; they were cut on an ordinary lathe by placing a reaming tool in the chuck and

TABLE 1
Miscellaneous Dimensions of the Rectification Apparatus

Distance between plates, in.....	8.5
Plate area, in. ²	28.891
Slot area per cap, in. ²	1.199
Slot area per plate, in. ²	2.398
Down-pipe area, in. ²	0.864
Vapor-riser area, in. ²	1.728
Heating surface, ft. ² :	
Top coil.....	4.82
Middle coil.....	3.85
Bottom coil.....	1.93
Total.....	10.60
Condenser:	
No. of tubes.....	43
Outside diameter of tubes, in.....	0.635
Length of tubes, in.....	17.75
Area of tubes, ft. ²	10.4
Area of tubes plus tube sheets, ft. ²	11.1

supporting the crucible on a special holder attached to the tool rest. This method made it possible to accurately control the spacing and the length of the slots.

A sheet metal baffle is placed between the two caps on each plate to direct the liquid across both caps before it passes to the down pipe. The down pipes, which are on alternate sides of the column on adjacent plates, and the vapor risers are made of 1-inch pipe, turned down to give a tight fit in holes drilled through the plate. The caps are bolted to the vapor risers with toggle bolts, the toggles of which straddle the lower ends of the risers. In order to provide sufficient space for the vapor, the lower ends of the vapor risers are cut away as shown in Figure 2.

TABLE 2
Bill of Materials for Bubble-Cap Rectification Column
 New York University

Quantity	Description	Size	Material
25 ft.	Std. pipe	2 in.	Black wrought steel
1½ ft.	" "	1½ in.	" " "
35 ft.	" "	1 in.	" " "
20 ft.	" "	¾ in.	" " "
70 ft.	" "	½ in.	" " "
100 ft.	" "	¾ in.	" " "
10 ft.	" "	½ in.	" " "
11	Std. nipple	6 in. x 8 in. long	" " "
1	" "	3 in. x 6 in. "	" " "
2	" "	3 in. x 3 in. "	" " "
3	" "	1 in. x 6 in. "	" " "
8	" "	1 in. x 3 in. "	" " "
5	" "	1 in. x 2 in. "	" " "
6	" "	1 in. x 1 in. "	" " "
2	" "	1 in., close	" " "
1	" "	¾ in. x 4 in. long	" " "
1	" "	¾ in. x 3 in. "	" " "
4	" "	¾ in. x 2 in. "	" " "
8	" "	¾ in. x 1 in. "	" " "
2	" "	¾ in., close	" " "
2	" "	½ in. x 3 in. long	" " "
7	" "	½ in. x 2 in. "	" " "
7	" "	½ in. x 1 in. "	" " "
7	" "	½ in., close	" " "
16	" "	¾ in. x 6 in. long	" " "
20	" "	¾ in. x 5 in. "	" " "
2	" "	¾ in. x 3 in. "	" " "
4	" "	¾ in. x 2 in. "	" " "
2	" "	¾ in. x 1 in. "	" " "
11	" "	¾ in., close	" " "
1	" "	½ in. x 2 in. long	" " "
4	" "	½ in. x 1 in. "	" " "
2	" "	½ in., close	" " "
24	" "	¾ in. x 4 in. long	" " "
12	" "	¾ in. x 2 in. "	" " "
12	" "	¾ in. x 1 in. "	" " "
2	Std. tee	2 in.	Black cast iron
1	" "	1 in.	" " "
11	" "	¾ in.	" " "
10	" "	½ in.	" " "
11	" "	¾ in.	" " "
1	" "	½ in.	" " "
2	" "	¾ x ¾ x ½ in.	" " "

TABLE 2—Continued

Quantity	Description	Size	Material
1	Std. tee	1½ x 1½ x ½ in.	Black cast iron
1	" "	1½ x 1½ x ½ in.	" " "
1	Std. 90 elbow	3 in.	" " "
4	" " "	2 in.	" " "
4	" " "	1 in.	" " "
10	" " "	¾ in.	" " "
24	" " "	½ in.	" " "
13	" " "	¾ in.	" " "
5	" " "	½ in.	" " "
7	Ground-joint union	¾ in.	Black malleable iron
10	" " "	½ in.	" " "
12	" " "	¾ in.	" " "
2	" " "	½ in.	" " "
1	" " "	½ in.	" " "
2	Bushing	1½ x ¾ in.	Black cast iron
2	"	1 in. x ¾ in.	Black malleable iron
3	"	¾ in. x ½ in.	" " "
3	"	¾ in. x ½ in.	Black cast iron
1	"	½ in. x ¾ in.	Black malleable iron
2	"	½ in. x ½ in.	" " "
2	Std. coupling	1 in.	" " "
2	" "	¾ in.	" " "
3	" "	½ in.	" " "
10	Std. cap	½ in.	" " "
1	Plug	1 in.	Black cast iron
12	"	¾ in.	" " "
6	"	¾ in.	Steel
36	"	½ in.	"
22	Screwed flange, 125 lb., faced and drilled	6 in.	Cast iron
2	Screwed flange, 125 lb., faced and drilled	3 in.	" "
2	Screwed flange, 125 lb., faced and drilled	2 in.	" "
1	Eccentric reducing screwed flange, 125 lb., faced and drilled	3 in. x 11 in.	" "
1	Reducing screwed flange, 125 lb., faced and drilled	2 in. x 11 in.	" "

TABLE 2—Continued

Quantity	Description	Size	Material
17	Std. return bend, close pattern, right and left hand threads	1 in.	Black malleable iron
1	Std. globe valve	$\frac{1}{2}$ in.	Brass
1	" " "	$\frac{1}{2}$ in.	"
4	Std. gate valve	1 in.	"
5	" " "	$\frac{1}{2}$ in.	"
9	" " "	$\frac{1}{2}$ in.	"
12	" " "	$\frac{1}{2}$ in.	"
2	" " "	$\frac{1}{2}$ in.	"
1	" " "	$\frac{1}{2}$ in.	"
30	Cock, tee handle	$\frac{1}{2}$ in. female pipe size x $\frac{1}{2}$ in. SAE male flared tube size	"
30	SAE flared tubing nuts, short-nosed	$\frac{1}{2}$ in.	"
150 ft.	Tubing	$\frac{1}{2}$ in. O.D. x 0.035 in. wall	Soft copper
1	Water gage	$\frac{1}{2}$ in.; 18 in. glass	Brass
1	Cylinder (ends ground square)	$6\frac{1}{2}$ in. O.D. x $\frac{1}{2}$ in. thick x 8 in. long	Pyrex glass
20	Crucible	50 cc.	Iron
5 ft.	Flats	$\frac{1}{2}$ in. x $1\frac{1}{2}$ in.	Cold-rolled steel
10	Sheet	$12 \times 12 \times \frac{1}{2}$ in.	" " "
1	"	$\frac{3}{8} \times 30 \times 63$ in.	" " "
3	"	$\frac{3}{8} \times 25 \times 25$ in.	" " "
100	Bolts and nuts	$\frac{1}{2}$ in. — 10 NC x 3 in. long	Steel
100	" " "	$\frac{1}{2}$ in. — 13 NC x 1 in. long	"
20	Toggle bolt	$\frac{1}{8}$ in. — 24 NS x 3 in. long	"
8	Expansion shield and bolt	$\frac{1}{2}$ in.	"
10 ft.	Hexagon stock	$\frac{3}{8}$ in.	Cold-rolled steel
6 ft.	Angles	$1\frac{1}{2}$ in.	Steel
16 ft.	"	2 in.	"
1	Sheet	$\frac{1}{4}$ in. x $4\frac{1}{2}$ ft. x 2 ft.	Galv. iron
2	Plate	$\frac{1}{4} \times 8 \times 8$ in.	Brass
3 ft.	Rod	$\frac{1}{2}$ in.	"
8	Wing nuts	$\frac{1}{2}$ in.	"
1	Pressure gage.	0 to 30 lb. press., 3 in. face	"

TABLE 2—Concluded

Quantity	Description	Size	Material
2	Sarco radiator steam trap, type H, angle pattern, 0 to 30 lb.	$\frac{1}{2}$ in.	Cast iron
1	Thermometer gage, angle form, model 221-D (Weston)	3 in. dial x 18 in. stem, 50 to 300°F.	Stainless steel stem
12 ft.	Pipe insulation	1 in.	85% Magnesia
12	" "	$\frac{3}{4}$ in.	" "
12	" "	$\frac{1}{2}$ in.	" "
24	" "	$\frac{1}{2}$ in.	" "
200 lb.	Insulating cement (No. 0352 Crane)	—	Asbestos

A thermometer well, turned from $\frac{1}{8}$ -inch hexagonal cold-rolled steel and threaded, enters each section, as shown in Figure 2, through a drilled and tapped $\frac{1}{2}$ -inch pipe plug placed in the wall of the section immediately above the plate.

Liquid and Vapor Samplers. The liquid samplers are $\frac{1}{2}$ -inch nipples, threaded into holes drilled and tapped through the flanges below the liquid levels on the plates; the vapor samplers are $\frac{1}{2}$ -inch nipples, threaded into holes 6 inches above the plates. All liquid and vapor samples are led individually through $\frac{1}{2}$ -inch copper tubing that passes through a water-cooled condenser box, which may be filled with ice if necessary.

Feed Pipes. The feed pipes, which are made from $\frac{1}{2}$ -inch pipe, nipples, and elbows, enter each section 4 inches above the plate and then turn down, forming a liquid seal on the plate between the down pipe from the plate above and the bubble cap nearest the down pipe. This is done to prevent splashing, which would affect the efficiency of the plate. The feed pipes are connected externally to two headers leading from the supply drum.

Vapor and Reflux Lines. The vapor from the top plate is led through 2-inch copper tubing to a copper condenser, with water inside the tubes and vapor outside. The condenser cools the overhead vapor to a cold liquid, which is returned either wholly or in part as reflux to the column. The cold reflux passes through an orifice weir*, and is then heated almost to its boiling point in a short, double-pipe, steam-heated exchanger before it enters the column at the top plate. It enters the

* The construction of this weir is shown in Figure 2 page 374.

plate in the same manner as the feed, and at the point where the down pipe would normally come from the plate above.

It is felt that cooling the reflux and then reheating it is the best practice, as experience with other student equipment has shown that difficulty is encountered with a condenser which removes only the latent heat from the vapors. With the arrangement described, the reflux can be heated and maintained within 1°F. of its boiling point without difficulty.

The Boiler. The boiler, the design of which is shown in Figure 3, has a shell made from $\frac{3}{8}$ -inch steel, rolled and welded into a cylinder 20 inches in diameter and 30 inches long. One end is closed by a plate welded to the cylinder, and the other end is provided with a welded flange, to which the cover plate is bolted. The boiler is mounted horizontally on four legs bolted to lugs welded to the sides.

Three sets of heating coils, made of 1-inch pipe and return bends, are mounted in the lower half of the boiler and welded to the cover plate. The top coil consists of ten pipes, the middle of eight, and the lower of four. Steam from a single header can be admitted to each coil individually or to any combination of them. Condensate from all three coils is removed through a single Sarco thermostatic steam trap.

Experience has shown that this design of heating surface works more satisfactorily than a jacketed kettle since, with the entire heating surface always submerged, no violent splashing of liquid occurs and superheating of vapor is avoided. In addition, the area of the heating surface may be varied at will.

The vapor leaves the boiler through a 3-inch standard pipe leading to the bottom of the column. The liquid returns from the column to the boiler through a 1-inch pipe that passes through the cover plate of the boiler; this line is provided with a liquid seal. A 1-inch pipe is welded into the top of the boiler for charging and for inserting a long-stemmed Weston dial thermometer, which extends below the liquid level; and a 1-inch pipe, for the withdrawal of distillation bottoms, is welded to the bottom of the boiler and passes to a water cooler. Gage glass fittings are welded to the rear plate.

Gasket and Insulation Materials. Asbestos sheeting was found to be a very satisfactory gasket material, and was used throughout. After assembly, the entire unit was insulated with 85 per cent magnesia pipe covering and asbestos cement.

Comments on Design. 1. The condenser and the vapor line from the column to the condenser were made of copper, since these parts were available at the time of construction. However, the entire equipment can be made of steel and cast iron.

2. Rotometers may be used in place of the weir box for measuring the quantity of reflux; this is particularly advantageous when part of the condensate is removed as product.

3. While automatic controls are desirable from the viewpoint of operation, it is felt that for instructional purposes manual control offers a greater opportunity for student experience.

Operation. The boiler is charged with the binary mixture that is to be rectified. The liquid and vapor sampling cocks and valves are closed, water is admitted to the condenser, the reflux valves are adjusted to provide the desired reflux ratio, and steam is then admitted to the selected coils of the boiler. When vapors appear at the top of the column, the steam pressure is adjusted to provide the desired distillation

TABLE 3
Specific Gravities of Carbon Tetrachloride-Toluene Mixtures

Per Cent CCl ₄ by weight	Specific gravity, d_{4}^{20}
0	0.8826
10	0.9055
20	0.9497
30	1.0000
40	1.0568
50	1.1196
60	1.1877
70	1.2702
80	1.3605
90	1.4663
100	1.5873

rate, the flow of the condenser water is adjusted to deliver the condensate at approximately room temperature, and steam is admitted to the reflux heater to heat the reflux to within 5° to 6°F. of the temperature at the top plate. When the temperatures have become constant, experimental data are taken.

Selection of the Binary Mixture. A mixture of carbon tetrachloride and toluene has been found to be ideal for student laboratory tests because: (1) the equilibrium curves show no abnormalities,⁸ and (2) analysis of the samples is simplified because the difference in densities of the two components is sufficiently great so that rather accurate and rapid determinations can be made with a Westphal balance.

Operation of the column with this mixture over a period of several months has shown that only about 1½ hours are required for equilibrium to be established, and only about 2 hours are required for sampling.

The data in the *International Critical Tables* for the density of the carbon tetrachloride-toluene system are apparently incorrect. The data in Table 3, which were supplied through the courtesy of Professor J. C. Elgin, Department of Chemical Engineering, *Princeton University*, are more accurate.

TYPICAL EXPERIMENT

Objects. To study the characteristics of a bubble-cap column operating under total reflux with a binary mixture, and to determine the plate efficiencies.

Auxiliary Equipment. To carry out this experiment, the following additional equipment is needed:

1. Thermometers:

Eight, 0 to 330°F.

Seven, 0 to 220°F.

Six, -30 to 120°F.

2. Westphal balance.

3. Sample bottles: 4-ounce capacity; 25 required.

4. Platform scales: capacity 250 pounds or greater.

5. Timer: One Eastman timer clock reading in seconds and minutes.

Procedure. 1. Charge the boiler with 20 gallons of a mixture of carbon tetrachloride and toluene, containing from 10 to 15 mol per cent of carbon tetrachloride.

2. Close all sampling cocks and valves, admit cooling water to the condenser, provide for proper drainage of steam condensate, adjust the reflux valves so that all of the overhead vapors return to the column as reflux, and then pass steam at 20 pounds per square inch gage pressure into the lowest two heaters of the boiler.

3. When vapors appear at the top of the column, adjust the steam pressure to give the desired distillation rate.

4. Adjust the cooling water rate so that the condensate is delivered at a temperature of 68 to 77°F.

5. Admit steam to the reflux heater until the temperature of the reflux is within 5 to 6°F. of the temperature of the top plate.

6. When the thermometers indicate that conditions have become constant, take and record the following data:

a. Pressure of the steam to the boiler.

b. Pressure of the steam to the reflux heater.

c. Temperature of:

(1) Liquor in the boiler.

(2) Liquor at every plate.

- (3) Condensate from the reflux heater.
 - (4) Water to and from the condenser.
 - (5) Reflux leaving the weir box, and reflux entering the column.
- d. Weight over a timed interval of:
- (1) Condensate from the boiler.
 - (2) Condenser water.

TABLE 4

Experimental Data and Calculated Results for the Distillation of a Mixture of Carbon Tetrachloride and Toluene

Conditions of operation: total reflux; atmospheric pressure.

Boiling points: carbon tetrachloride, 170.2°F.; toluene, 231.1°F.

Heating coils used: middle and lower.

Steam: 15 lb./in.² gage; 21.48 lb./hr.

Reflux rate: 182 lb./hr.

Condenser water: in at 43°F.; out at 46°F.; rate, 6,630 lb./hr.

Plate No. from top	Temp., °F.	Specific gravity		Composition, mol fraction CCl ₄		Murphree plate efficiency, per cent
		Liquid, d_{40}^{25}	Con- densed vapor, d_{40}^{25}	Liquid	Vapor	
1	171	1.5500	1.5866	0.950 (L_1)	0.995 (V_1)	161
2	172	1.5290	1.5528	0.920 (L_2)	0.952 (V_2)	71.1
3	173	1.4930	1.5260	0.873 (L_3)	0.916 (V_3)	61.5
4	175	1.4470	1.4940	0.811 (L_4)	0.873 (V_4)	60.8
5	178	1.3940	1.4425	0.740 (L_5)	0.805 (V_5)	49.2
6	183	1.3190	1.3888	0.640 (L_6)	0.732 (V_6)	55.0
7	192	1.1999	1.3120	0.483 (L_7)	0.632 (V_7)	70.9
8	200	1.0950	1.1840	0.340 (L_8)	0.463 (V_8)	59.7
9	212	0.9924	1.0780	0.191 (L_9)	0.314 (V_9)	76.4
10	220	0.9420	0.9920	0.121 (L_{10})	0.192 (V_{10})	57.3
Boiler.....	226	0.8758	0.9190	—	—	—
Reflux at orifice.....	56	1.5820	—	0.992 (L_R)	—	—
Reflux into column.....	169	1.5820	—	0.992 (L_R)	—	—
Liquid return to boiler...	—	0.8918	—	0.050 (L_s)	—	—

Overall plate efficiency: 78%.

Heat: Given up by steam, 20,300 B.t.u./hr.

Removed by cooling water, 19,900 B.t.u./hr.

Unaccounted for, 400 B.t.u./hr. = 1.97%.

7. Take 100-ml. samples of liquid and condensed vapor from each plate, from the boiler, and from the top of the column, and determine their specific gravities, at 25°C. referred to water at 4°C., by means of a Westphal balance.

In obtaining these samples, start at the top of the column and work downward, taking a vapor sample first, and then a liquid sample. The

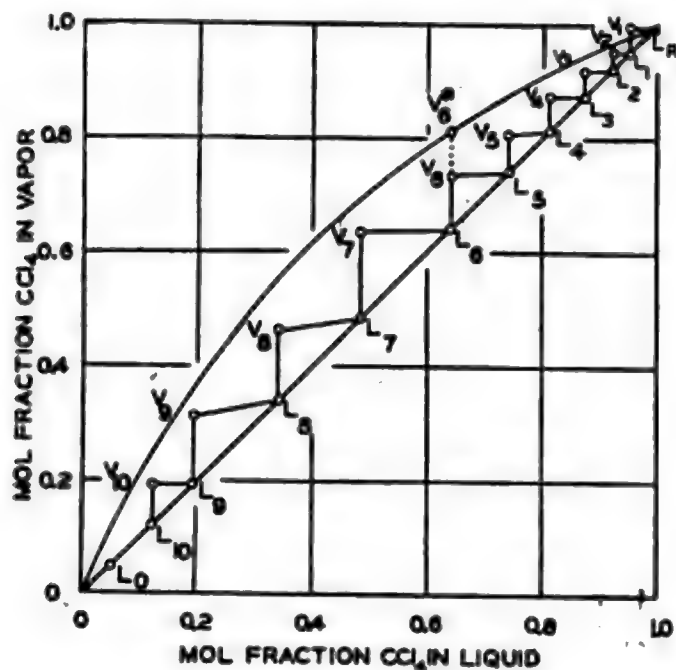


Figure 4. McCabe-Thiele Analysis for Distillation of a Mixture of Carbon Tetrachloride and Toluene at Total Reflux

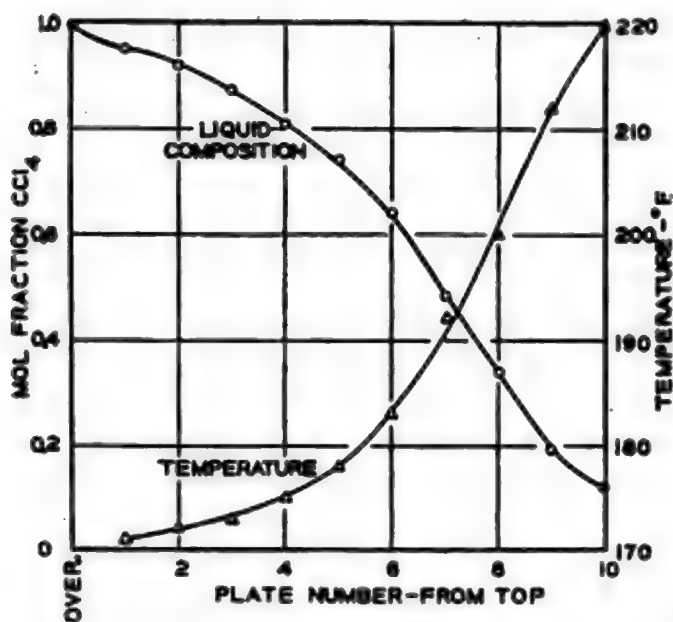


Figure 5. Temperatures and Compositions of the Liquid on the Plates for Distillation of a Mixture of Carbon Tetrachloride and Toluene at Total Reflux

samples should be withdrawn slowly, making certain that the plates are not drained; and a few minutes should be allowed between successive

samples. The samples should be placed immediately in suitably stoppered, labeled bottles.

8. From the experimental data, calculate:

- a. The theoretical number of plates and the overall efficiency by the McCabe-Thiele graphical method.^{1, 4}
- b. The Murphree efficiency for each plate.^{2, 6}
- c. The heat balance over the entire system.

9. On a single graph, plot the temperature and the liquid composition on each plate against the plate number.

Experimental Data and Calculated Results. The experimental data and calculated results for the run are given in Table 4; the McCabe-Thiele analysis of the data is shown in Figure 4; and the temperatures and liquid compositions on the plates are shown in Figure 5.

Sample Calculation of the Murphree Plate Efficiency. The Murphree plate efficiency is calculated by means of the equation:

$$E = \frac{y_n - y_{n+1}}{y_n^* - y_{n+1}} \times 100$$

For example, the calculation for plate No. 6 is

$$E = \frac{V_6 - V_7}{V_6^* - V_7} \times 100 = \frac{0.732 - 0.632}{0.814 - 0.632} \times 100 = 55.0$$

where

E = Murphree plate efficiency, per cent.

and, from Figure 4:

$y_n = V_6 = 0.732$ = mol fraction of carbon tetrachloride in the vapor leaving plate No. 6.

$y_{n+1} = V_7 = 0.632$ = mol fraction of carbon tetrachloride in the vapor rising to plate No. 6.

$y_n^* = V_6^* = 0.814$ = mol fraction of carbon tetrachloride in the vapor in equilibrium with the liquid leaving plate No. 6.

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A PACKED DISTILLATION COLUMN

Designed and Constructed by
The School of Chemical Engineering
Cornell University*

Description. This packed distillation unit, which is shown in Figure 1, includes a column, a steam-heated still, and a condenser. The column is made from a 12½-foot length of 3-inch pipe, and it has a total packed height of 12.33 feet. The still is made from a flanged 1½-foot length of 12-inch pipe, and has six ½-inch heating pipes, which provide a heating surface of approximately 2 square feet. The condenser is 3 feet long, and has twenty-seven ½-inch I.D. tubes, arranged vertically in a single pass.

The equipment is constructed almost entirely of standard pipe and fittings, and it is made so as to permit easy removal of the packing.

Materials and Cost. The materials used in constructing this distillation unit, exclusive of the condenser, are listed in Table 1. The weir box was made in the shops of *Cornell University* at a cost of \$30; the rest of the items cost about \$90. The materials for the condenser are not included in the bill of materials and its dimensions are not shown in the drawing because the condenser was not designed for this column and is considerably oversize.

Notes on Construction. 1. To facilitate removal of the packing, line *L*, for the return of liquid to the still, is connected to the bottom of the column through a flange union.

2. The packing is supported on a perforated plate that rests on a tripod which, in turn, rests on a bushing in the lower half of the flange.

3. A ¾-inch steam line is provided in vapor line *V*, leading to the condenser, to prevent fractional condensation of the vapors.

4. The quantity of reflux is measured by means of a weir box in the return line from the condenser. The construction of this weir box is shown in Figure 2.

5. The column is insulated with 85 per cent magnesia. Differential thermocouples are provided across the thickness of the insulation at 3 points along the length of the column, so that the temperature drop across the insulation can be measured and the heat loss from the column can be calculated.

* Reported by C. C. Winding and F. H. Rhodes.

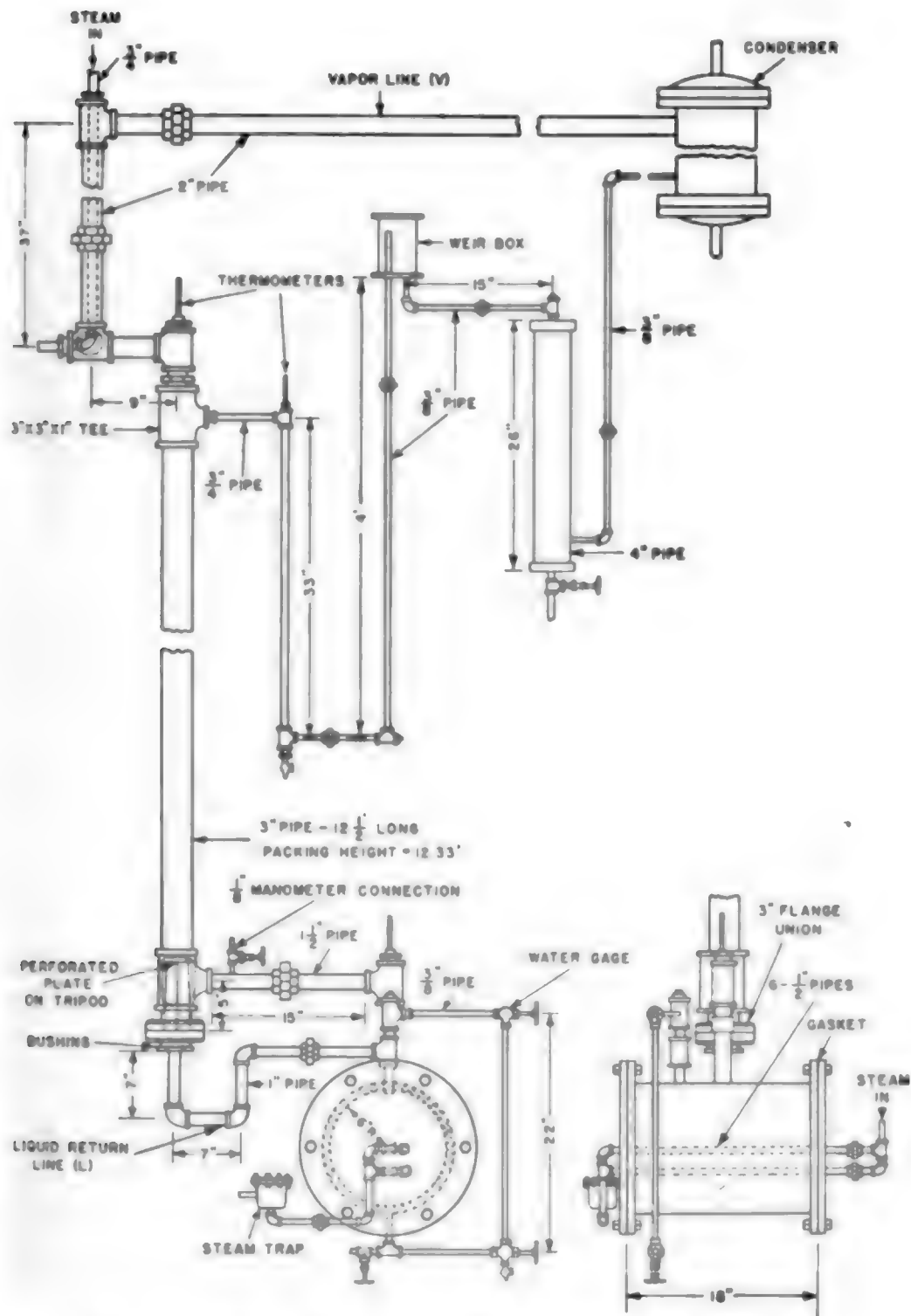


Figure 1. Packed Distillation Column—Cornell University

TABLE 1
Bill of Materials for Packed Distillation Column
 Cornell University

Quantity	Description	Size	Material
1½ ft.	Std. pipe	12 in.	Black wrought steel
2 ft. 2 in.	" "	4 in.	" " "
12½ ft.	" "	3 in.	" " "
3 ft.	" "	2 in.	" " "
6 ft.	" "	¾ in.	" " "
12 ft.	" "	½ in.	" " "
5½ ft.	" "	½ in.	" " "
1	Std. nipple	3 in., close	" " "
2	" "	2 in. x 7 in. long	" " "
1	" "	2 in., close	" " "
2	" "	1½ in. x 7 in. long	" " "
1	" "	1½ in. x 6 in. "	" " "
1	" "	1 in. x 7 in. "	" " "
3	" "	1 in. x 6 in. "	" " "
2	" "	1 in. x 4 in. "	" " "
2	" "	1 in. x 2 in. "	" " "
1	" "	¾ in. x 8 in. "	" " "
3	" "	½ in. x 6 in. "	" " "
1	" "	½ in. x 3 in. "	" " "
16	" "	½ in. x 2 in. "	" " "
2	" "	½ in. x 10 in. "	" " "
2	" "	½ in. x 7 in. "	" " "
4	" "	½ in. x 4 in. "	" " "
1	" "	½ in. x 3 in. "	" " "
1	" "	½ in., short	" " "
1	" "	½ in. x 4 in. long	" " "
1	Std. tee	3 x 3 x 1½ in.	Black malleable iron
1	" "	3 x 3 x 1 in.	" " "
3	" "	2 in.	" " "
1	" "	1½ in.	" " "
1	" "	1 in.	" " "
1	" "	1 x 1 x ½ in.	" " "
1	" "	¾ in.	" " "
1	" "	¾ x ¾ x ½ in.	" " "
6	" "	½ in.	" " "
3	" "	½ in.	" " "
3	Std. 90° elbow	1 in.	" " "
1	" " "	¾ in.	" " "
8	" " "	½ in.	" " "
2	" " "	½ in.	" " "
2	Ground-joint union	2 in.	" " "
1	" " "	1½ in.	" " "
1	" " "	1 in.	" " "

TABLE 1—Concluded

Quantity	Description	Size	Material
7	Ground-joint union	$\frac{1}{2}$ in.	Black malleable iron
4	" " "	$\frac{1}{2}$ in.	" " "
1	Bushing	3 x 2 in.	Cast iron
1	"	3 x 1 in.	" "
2	"	2 x $\frac{3}{4}$ in.	" "
1	"	2 x $\frac{1}{2}$ in.	" "
1	"	1 $\frac{1}{2}$ x $\frac{1}{2}$ in.	" "
1	"	1 x $\frac{3}{4}$ in.	" "
2	"	$\frac{3}{4}$ x $\frac{1}{2}$ in.	" "
2	Std. cap	4 in.	Black malleable iron
1	Plug	1 in.	Cast iron
2	"	$\frac{1}{2}$ in.	Steel
1	Std. flange union, gasket type	3 in.	Cast iron
1	Std. gate valve	$\frac{1}{2}$ in.	Brass
1	" " "	$\frac{1}{2}$ in.	"
1	Std. globe valve	$\frac{1}{2}$ in.	"
1	Pet cock	$\frac{1}{2}$ in.	"
2	Gasket	15 in. O.D. x $\frac{1}{8}$ in. thick	Asbestos composition
1	"	6 $\frac{1}{2}$ in. O.D. x $\frac{1}{8}$ in. thick	" "
1	Steam trap	$\frac{1}{2}$ in.	Cast iron
1	Water gage	$\frac{1}{2}$ in. with 21 in. glass	Brass
1	Weir box*	—	—
1	Tripod	3 in. O.D. x 7 in. high	Steel
1	Perforated plate	3 in. diam. x $\frac{1}{8}$ in. thick	"
$\frac{1}{2}$ cu. ft.	Marbles	$\frac{1}{2}$ in. diam.	Glass
$\frac{1}{2}$ cu. ft.	Raschig rings	$\frac{1}{2}$ in. diam. x $\frac{1}{2}$ in. long	Brass
3	Thermometer	-10 to 120°C.	Mercury
1 $\frac{1}{2}$ ft.	Pipe insulation	12 in.	85% magnesia
12 $\frac{1}{2}$ ft.	" "	3 in.	" "
6 ft.	" "	2 in.	" "
1 ft.	" "	1 $\frac{1}{2}$ in.	" "
2 ft.	" "	1 in.	" "

* See Figure 2.

Comments on Design. 1. The assembly of this unit, as shown in Figure 1, provides for operation only with infinite reflux. However, provision can easily be made for the removal of product to permit operation at various reflux ratios.

2. Two types of packing are used at *Cornell University*:

a. *Raschig rings*. These are $\frac{3}{8}$ -inch lengths of brass pipe, $\frac{1}{8}$ -inch I.D. and $\frac{1}{4}$ -inch O.D. When the column is packed to a height of 12.33 feet, this packing provides a total surface, including that of the walls, of 5.16 square feet, and a free space of 78 per cent.

b. *Glass marbles*. These are approximately $\frac{3}{8}$ -inch in diameter and, with a packed height of 12.33 feet, they provide a total surface, including that of the walls, of 4.01 square feet. The free space with this packing is 36 per cent.

Although the size of these packings is rather large compared to the diameter of the column, they are readily available at a reasonable cost.

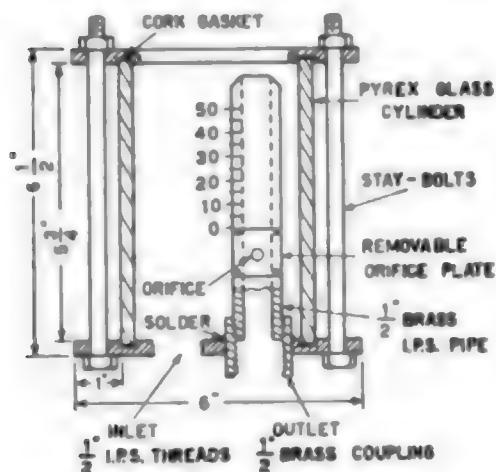


Figure 2. Details of Construction of the Wier Box

3. The still may be designed to permit easy removal of the heating element by using return bends and bringing both the steam inlet and the condensate outlet through the same flanged cover. This construction is shown in Figure 3 on page 344.

Operation. At *Cornell University*, benzene-toluene and ethanol-water are the usual binary mixtures used in student distillation experiments. With either mixture, it is desirable that the concentration be such that the theoretical steps of the McCabe-

Thiele diagram appear on that section of the graph that has a rather wide spread between the equilibrium curve and the operating line.

In making a run with benzene-toluene, the still is first charged with toluene, the reading on the gage glass is noted, and the toluene is heated to boiling. When vapors appear at the top of the column, benzene is slowly run into the still until the level of the liquid is restored to its original value. A similar procedure is followed in making a run with ethanol-water; in this case, the still is first charged with water.

When equilibrium conditions have become established, the temperatures at various points in the system and the rate of distillation are measured, and the composition of the distillate and the still contents are determined.

TYPICAL EXPERIMENT

Object. To determine the column height equivalent to a theoretical plate for the distillation of: (1) benzene-toluene, and (2) ethanol-water,

at different vapor velocities and with two packing materials—brass Raschig rings and glass marbles.

Procedure. 1. Charge the still with toluene until it is about two-thirds full, note the height on the gage glass, and then turn on the steam. When toluene vapors appear at the top of the column, add

TABLE 2
Experimental Data and Calculated Results for the Distillation of Benzene-Toluene and Ethanol-Water Mixtures

Item No.	System.....	Benzene-Toluene				Ethanol-Water	
	Type of Packing.....	Raschig Rings		Marbles		Marbles	
		Run Number					
		1	2	3	4	5	6
1	Packed height, ft.....✓	12.33	12.33	12.33	12.33	12.33	12.33
2	Conc. of still contents, per cent more volatile constituent.....	13.5	23.9	19.8	14.2	0.4	0.58
3	Conc. of reflux, per cent more volatile constituent.....	99.6	99.7	99.8	98.3	76.8	83.3
4	Reflux, gal./hr.....✓	2.04	3.71	2.06	3.96	—	—
5	Temp. at top of column, °C.....	80.3	80.6	80.0	80.1	80.2	77.3
6	Temp. of reflux, °C.....	16.3	12.1	15.2	12.0	47.9	11.8
7	Temp. drop of reflux, °C.....	64.0	68.5	64.8	66.1	41.3	65.8
8	Temp. drop of reflux, °F.....	115.2	123.3	116.6	122.6	74.3	118.4
9	Reflux, lb./hr.....✓	14.85	27.0	14.96	28.5	13.4	14.17
10	Specific heat of reflux, B.t.u./(lb. × °F.).....✓	0.47	0.47	0.47	0.47	0.68	0.65
11	Sensible heat loss of reflux, B.t.u./hr.....	804	1,560	820	1,650	877	1,060
12	Latent heat of reflux, B.t.u./lb.....✓	169.4	169.4	169.4	169.4	450	425
13	Vapors condensed by reflux, lb./hr.....	4.75	9.22	4.85	9.76	1.80	2.57
14	Vapors condensed by reflux, mols/hr.....	0.0608	0.118	0.0622	0.125	0.038	0.062
15	Vapors condensed through radiation, mols/hr.....	0.037	0.037	0.037	0.037	0.22	0.22
16	Vapor in column, mols/hr. (av.).....	0.270	0.483	0.273	0.510	0.300	0.416
17	Temp. of the vapor, °C. (est. av.).....✓	90	90	90	90	95	95
18	Vapor in column, ft ³ /hr. (av.).....	129	230	130	243	139	202
19	Velocity of vapor through free space, ft./sec.....	0.86	1.62	1.95	3.65	2.84	3.04
20	Number of theoretical plates.....	7.72	7.3	8.0	6.13	6.6	8+
21	Height of theoretical plate, ft.....	1.59	1.69	1.54	2.01	1.87	1.54
22	Flooding rate, gal./hr.....	—	—	—	—	8.0	9.5
23	Vapor velocity through free space at flooding point, ft./sec.....	3.5	3.5	4.5	4.5	—	—

sufficient benzene to the still to bring the liquid level to the mark previously noted.

2. Adjust the steam pressure so that distillation occurs at a rate of about 2 to 4 gallons per hour, read the thermometers at the top and at the bottom until they indicate that the system has reached equilibrium, and then take the following data:

a. The temperatures of: (1) the vapor entering the column, (2) the vapor leaving the column, and (3) the reflux.

- b. The height of liquid in the weir box.
- c. The temperature drop through the insulation.
3. Withdraw samples of liquid from the still and the reflux line, and:
 - (a) measure their specific gravities by means of a hydrometer, and (b) determine their compositions by careful distillation in a small laboratory distillation column.
4. Repeat the experiment at different vapor velocities and with a different packing material, and then remove the benzene-toluene mixture from both the column and the still and make another series of runs with an ethanol-water mixture.

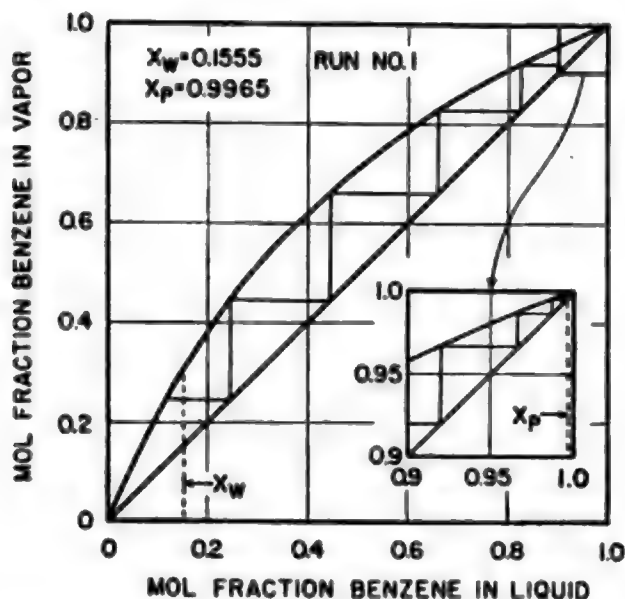


Figure 3. McCabe-Thiele Diagram for Distillation of a Mixture of Benzene-Toluene at Total Reflux

5. Calculate for each run:
 - a. The quantity of reflux in the column. This is comprised of three parts: (1) the quantity of liquor returned from the condenser, (2) the condensation of vapors in the column by this cold liquor, and (3) the condensation of vapors in the column through loss of heat by radiation.
 - b. The vapor velocity through the free space.
 - c. The theoretical number of plates in the column.
 - d. The height equivalent to a theoretical plate.

Typical Data from a Report. A summary of the experimental data and the calculated results for several runs are given in Table 2, and the McCabe-Thiele diagram for Run No. 1 is shown in Figure 3. The method used in treating the data is illustrated by the calculations for one run.

Calculations for Run No. 1.Items 7 and 8*: *Temperature drop of reflux.*

Temp. at top of column, °C.	= 80.3	Item 5
Temp. of reflux, °C.	= 16.3	Item 6
$80.3 - 16.3 = 64.0^{\circ}\text{C.}$		Item 7
$64.0 \times 1.8 = 115.2^{\circ}\text{F.}$		Item 8

Item 9: *Quantity of reflux.*

Gallons of reflux per hour	= 2.04	Item 4
Specific gravity	= 0.873	
One gallon	= 231 in. ³	
One cubic foot	= 1728 in. ³	
Density of water	= 62.4 lb./ft. ³	
$2.04 \times (231/1728) \times 0.873 \times 62.4$	= 14.85 lb./hr.	

Item 11: *Sensible heat loss of reflux.*

Reflux, lb./hr.	= 14.85	Item 9
Temp. drop of reflux, °F.	= 115.2	Item 8
Specific heat of reflux, ⁵		
B.t.u./(lb. \times °F.)	= 0.47	Item 10
$14.85 \times 115.2 \times 0.47$	= 804 B.t.u./hr.	

Item 14: *Vapors condensed by reflux.*

Sensible heat loss of reflux, B.t.u./hr.	= 804	Item 11
Latent heat of reflux, ⁴ B.t.u./lb.	= 169.4	Item 12
Average molecular weight of reflux		
$= 0.996 \times 78.11 + 0.004 \times 92.13$	= 78.17	
$(804/169.4) \times (1/78.17)$	= 0.0608 mols/hr.	

Item 15: *Vapors condensed through radiation, mols/hr.*

$$= \frac{\text{heat loss through radiation, B.t.u./hr.}}{\text{latent heat, B.t.u./mol}}$$

INSULATION:

- a. Outside diameter, in. = 5.5
- b. Inside diameter, in. = 3.5
- c. Height, ft. = 12.5
- d. Thickness, ft. = $1/12 = 0.0834 = L$
- e. Mean area, ft.²

$$= \frac{5.5 - 3.5}{2.303 \log_{10} (5.5/3.5)} \times \frac{\pi}{12} \times 12.5 = 14.45 = A$$

* Item numbers refer to quantities in Table 2.

$$f. \text{ Mean temp. drop across insulation } \backslash \\ = 39^{\circ}\text{C. (71.2}^{\circ}\text{F.)}^* = \Delta t$$

$$g. \text{ Thermal conductivity}^2, \\ \text{B.t.u.-ft.}/(\text{hr.} \times \text{ft.}^2 \times ^{\circ}\text{F.}) = 0.04 = k$$

REFLUX:

$$a. \text{ Average molecular weight} = 78.17$$

$$b. \text{ Latent heat, B.t.u./lb.} = 169.4$$

The heat lost by radiation is¹

$$\frac{kA\Delta t}{L} = \frac{0.04 \times 14.45 \times 71.2}{0.0834} = 494 \text{ B.t.u./hr.}$$

and the quantity of vapors condensed through radiation is $494/(169.4 \times 78.17) = 0.037 \text{ mols/hr.}$

Item 16: *Vapor in column, (average mols/hr.)*

= average mols vapor at midsection of column

= mols reflux + mols vapor condensed by reflux
+ mols vapor condensed through radiation in
upper half of column

$$= \frac{\text{Item 9}}{\text{av. mol. wt. of reflux}} + \text{Item 14} + \frac{1}{2} (\text{Item 15})$$

$$= (14.85/78.17) + 0.0608 + \frac{1}{2}(0.037)$$

$$= 0.270 \text{ mols/hr.}$$

Item 18: *Vapor in column, (average ft.³/hr.).*

$$\text{Temp. of vapor (est. av.)} = 90^{\circ}\text{C.} \dots \dots \text{Item 17}$$

$$\text{Mols of vapor per hr.} = 0.270 \dots \dots \text{Item 16}$$

$$0.270 \times 359 \times \frac{90 + 273}{273} = 129 \text{ ft.}^3/\text{hr.}$$

Item 19: *Velocity of vapor through free space.*

$$\text{Cross-sectional area of column, in.}^2 = 7.393$$

$$\text{Per cent free space} = 78$$

$$\frac{129}{7.393/144} \times \frac{1}{3600} \times \frac{100}{78} = 0.89 \text{ ft./sec.}$$

Item 20: *Number of theoretical plates.*

$$\text{Composition of reflux} = 99.6\% \text{ benzene} \dots \text{Item 3}$$

$$\text{Composition of still contents}$$

$$= 13.5\% \text{ benzene} \dots \dots \dots \text{Item 2}$$

* Determined by differential thermocouples located at three points along the column.

Molecular weight of benzene = 78.11

Molecular weight of toluene = 92.13

Mol fraction of benzene in reflux

$$= \frac{99.6/78.11}{99.6/78.11 + 0.4/92.13} = 0.9965 = x_p$$

Mol fraction of benzene in still contents

$$= \frac{13.5/78.11}{13.5/78.11 + 86.5/92.13} = 0.155 = x_w$$

These values are plotted on the McCabe-Thiele diagram,¹ Figure 3; the number of theoretical plates is 7.72.

Item 21: *Height of a theoretical plate.*

Height of packed section, ft. = 12.33

$$12.33/7.72 = 1.59 \text{ ft.}$$

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SECTION 6

GAS ABSORPTION

INTRODUCTION

Gas absorption, in general, involves the treatment of a gaseous mixture with a liquid which dissolves one or more of the gaseous constituents. In most cases of importance, the gas phase is composed of two constituents, one of which is soluble while the other is insoluble, or inert. Industrially, most absorption processes are carried out in towers in which the gas is passed upward countercurrent to the flow of liquid, because countercurrent operation is essential for maximum absorption with a minimum quantity of liquid. To obtain intimate contact between the liquid and the gas, the towers are generally of the packed or bubble-cap type similar to those used for distillation, although for some purposes, such as the recovery of ammonia from coal gas, spray towers are used. However, where the gas is very soluble and its heat of solution is high, as, for example, in the manufacture of hydrochloric acid by the absorption of hydrogen chloride in water, intimate contact may be undesirable. In such cases, the efficiency of absorption is determined principally by the effectiveness of the heat removal, and, therefore, cooling surface is generally more important than absorbing surface.

Mechanism of Gas Absorption. When a mixture of a soluble gas *A* and an insoluble gas *B* is brought into contact with a solvent liquid *C*, the soluble gas must diffuse in series through a film of the inert gas and a film of the liquid before it can dissolve in the main body of liquid.¹⁸ Therefore, the rate of absorption, after conditions have become constant, is equal to the rate of diffusion through either film, and this may be expressed as¹²

$$N_A = \frac{D_G P A (p_{AG} - p_{Ai})}{RT B_G p_{BM}} = k_G A (p_{AG} - p_{Ai}) \quad (6.1)$$

$$= \frac{D_L (C_{AL} + C_{CL})(A)(C_{Ai} - C_{AL})}{B_L C_{CM}} = k_L A (C_{Ai} - C_{AL}) \quad (6.2)$$

where

N_A = rate of diffusion, lb. mols/hr.

D_G = diffusivity of component *A* through the gas film, ft.²/hr.

P = total pressure, atm.

A = interfacial, or absorbing, area, ft.²

- R = gas constant, ft.³-atm./(lb. mol. \times $^{\circ}R$).
 T = absolute temperature, $^{\circ}R$.
 B_G = equivalent thickness of the gas film, ft.
 p_{AG} = partial pressure of component A in the gas phase, atm.
 p_{Ai} = partial pressure of component A at the gas-liquid interface, atm.
 p_{BM} = log-mean partial pressure of inert component B in the gas film, atm.
 k_G = gas film coefficient, lb. mols of A absorbed per hour per ft.² per atm. partial pressure difference.
 D_L = diffusivity of component A through the liquid film, ft.²/hr.
 C_{AL} = concentration of component A in the liquid phase, lb. mols/ft.³
 C_{CL} = concentration of the solvent C in the liquid phase, lb. mols/ft.³
 k_L = liquid-film coefficient, lb. mols of A absorbed per hour per sq. ft. per unit concentration difference in lb. mols/ft.³

In order to calculate the rate of absorption by means of Equations (6.1) and (6.2), it is necessary to know the values of k_G and k_L and the equilibrium relationship between the partial pressure p of the gas and its concentration C in the solution.

Equilibrium Relationships. The solubility of a gas in a liquid depends upon the temperature, the partial pressure of the gas, and the total pressure on the system. If the total pressure is not too great—less than about 5 atmospheres—its effect is usually negligible,² but the solubility is greatly decreased by rise in temperature. Although the equilibrium solubility relationships for any actual system can only be determined experimentally, it is sometimes sufficiently accurate to assume that the system is ideal, in which case the relationship is expressed by Raoult's law:

$$p = P' \times MF \quad (6.3)$$

where

P' = vapor pressure of the pure component, atm.

MF = mol fraction of the component in the liquid.

In many other cases, even though Raoult's law does not apply, the solubility relationship is expressed by Henry's law

$$p = HC \quad (6.4)$$

where H is the experimentally determined Henry's law constant.

Film Coefficients. The magnitude of a film coefficient depends upon the particular system under consideration, the design of the equipment, and upon the conditions of temperature, pressure, concentration, and gas and liquor rates. Therefore, even though data may be available for

a particular system, it frequently becomes necessary to estimate values of the coefficients for conditions other than those under which they were determined; and, if no data are available, it may be necessary to estimate the coefficients for one system from the data for another. This requires considerable judgement, but it is facilitated by the use of certain relationships.

The magnitudes of the film coefficients are determined, principally, by the magnitudes of the diffusivities and the equivalent film thicknesses, for it is evident from Equations (6.1) and (6.2) that

$$k_g = \frac{D_g P}{RT B_g p_{BM}} \quad (6.5)$$

and

$$k_L = \frac{D_L (C_{AL} + C_{CL})}{B_L C_{LM}} \quad (6.6)$$

Experimental values of gas and liquid diffusivities are available for a number of systems,^{7, 13} and in the absence of experimental data they may be estimated from the equations^{1, 6}

$$D_g = 0.0166 \frac{(T')^{3/2}}{P[(V'_A)^{1/3} + (V'_B)^{1/3}]^2} \left(\frac{1}{M_A} + \frac{1}{M_B} \right)^{1/2} \quad (6.7)$$

and

$$D_L = \frac{B^* \left(\frac{1}{M_A} + \frac{1}{M_C} \right)^{1/2}}{A_A^* A_C^* \mu_c^{1/2} [(V'_A)^{1/3} + (V'_C)^{1/3}]^2} \quad (6.8)$$

where

D_g and D_L = gas and liquid diffusivities, respectively, ft.²/hr.

T = absolute temperature, °K.

M_A , M_B and M_C = molecular weight of the solute A , the inert gas B , and the solvent liquid C , respectively.

P = total pressure, atm.

V'_A , V'_B and V'_C = molecular volume of the components A , B , and C , respectively.

μ = viscosity of the solvent.

A_A^* , A_C^* and B^* = constants.

On the other hand, data for equivalent film thicknesses are practically non-existent for the actual conditions under which absorption is carried out. This is due to the fact that film thicknesses depend, in large part,

on the mass velocities of the fluids and the direction of flow; and the flow of fluids in an absorption column is extremely complex.

For wetted-wall towers, equivalent gas-film thicknesses have been correlated by the following equation, in which all of the groups are dimensionless¹⁶

$$\frac{d}{B_g} = 0.023 \left(\frac{du\rho}{\mu} \right)^{0.83} \left(\frac{\mu}{\rho D_g} \right)^{0.44} \quad (6.9)$$

where

d = tower diameter.

B_g = gas-film thickness.

u = gas velocity.

ρ = gas density.

μ = gas viscosity.

D_g = diffusivity.

Although this equation holds only for the special case where the absorbing liquid is flowing as a continuous film down the tube wall, it provides a means for estimating the probable effects of changes in the variables upon the effective film thickness in other types of towers.

For estimating liquid-film coefficients in packed towers, the equation of Sherwood and Holloway is very useful^{11, 16}

$$\frac{k_L a}{D_L} = \alpha' \left(\frac{L}{A' \mu} \right)^{1-n'} \left(\frac{\mu}{\rho D_L} \right)^{0.5} \quad (6.10)$$

where

a = effective absorbing area per unit volume, ft.²/ft.³

L = liquor rate, lb. mols/hr.

A' = cross-sectional area, ft.²

μ = viscosity of the liquid, lb./(ft. \times hr.).

ρ = density, lb./ft.³

D_L = diffusivity, ft.²/hr.

α' and n' = constants, whose values depend upon the type of packing, although for most packing materials, n' is approximately 0.25.

A similar relationship is not available for the gas-film coefficient, but for a given system, k_g varies approximately as the 0.8 power of the gas rate.

Overall Coefficients. Although, from a fundamental point of view film coefficients are to be preferred to overall coefficients, it is usually

much more convenient to employ overall coefficients and to write the absorption equation as

$$N_A = K_G A (p_{AG} - p_{AL}) = K_L A (C_{AG} - C_{AL}) \quad (6.11)$$

where

p_{AL} = the equilibrium partial pressure corresponding to concentration C_{AL} , atm.

C_{AG} = the equilibrium concentration corresponding to partial pressure p_{AG} , lb. mols/ft.³

K_G = the overall absorption coefficient, lb. mols/(hr. \times ft.² \times atm. overall partial pressure difference).

K_L = the overall absorption coefficient, lb. mols/(hr. \times ft.² \times unit overall concentration difference in mols per cu. ft.).

Overall coefficients, however, should only be employed for systems where Henry's law applies over the concentration range involved, for only in this case will the ratio of K_L to K_G remain constant.

Volumetric Coefficients. When absorption takes place in packed towers, it is difficult, and frequently impossible, to estimate the effective absorbing area. In such cases, a knowledge of coefficients based on area is of little help. Therefore, it is common practice to express absorption coefficients in terms of volume rather than area. This is done by replacing the area term A in the rate equations by the equivalent term aV , where a is the effective area per cubic foot of tower volume, and then treating the products $k_G a$, $K_G a$, $k_L a$, and $K_L a$ as individual terms. For example, in the equation

$$N_A = K_G a V (p_{AG} - p_{AL}) \quad (6.12)$$

$K_G a$ is the overall coefficient expressed as mols per hour per cubic foot of packed volume per atmosphere partial pressure difference, and V is the packed volume in cubic feet.

ABSORPTION EQUIPMENT DESIGN

Equations (6.1), (6.2), (6.11), and (6.12) apply only where conditions of partial pressure and concentration are constant. In an actual column, however, conditions vary from top to bottom. This is shown in Figures 1, 2, and 3. Figure 1 is a diagrammatic sketch of conditions in a countercurrent absorption tower; Figure 2 shows the conditions in the tower plotted in terms of partial pressure and concentration; and Figure 3 shows the conditions plotted in terms of mol ratios.

The equilibrium curves vary in shape depending upon the particular

system, but for the special case where Henry's law applies, the equilibrium curve on the pressure-concentration diagram is a straight line through the origin, while on the mol ratio diagram it is a rectangular hyperbola.

The particular value of the mol ratio diagram is due to the fact that in mol ratio units the operating line is straight, while in pressure-concentration units it is a curve. This is made evident by writing a

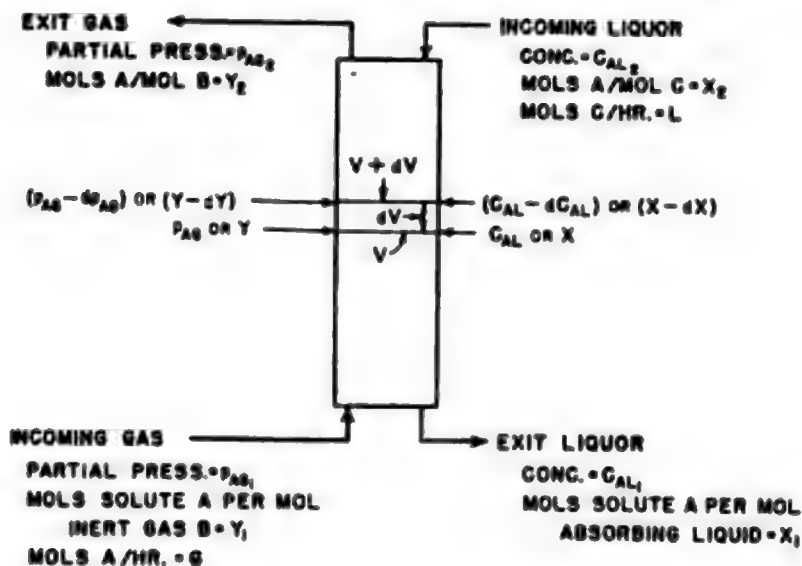


Figure 1. Diagrammatic Sketch of Conditions in a Countercurrent Absorption Tower

material balance between the conditions at the bottom of the tower— (X_1, Y_1) and (C_{AL1}, p_{A1}) —and at any point within the tower— (X, Y) and (C_{AL}, p_{A1}) :

$$L(X_1 - X) = G(Y_1 - Y) \quad (6.13)$$

or

$$L \left(\frac{C_{AL1}}{p_1 - C_{AL1}} - \frac{C_{AL}}{p - C_{AL}} \right) = G \left(\frac{p_{A1}}{P - p_{A1}} - \frac{p_{A1}}{P - p_{A1}} \right) \quad (6.14)$$

where

L = lb. mols of absorbing liquid per hr.

G = lb. mols of inert gas per hr.

X = lb. mols of absorbable constituent per mol of absorbing liquid.

Y = lb. mols of absorbable constituent per mol of inert gas.

C_{AL} = concentration of the absorbable constituent in the liquid, lb. mols/ft.³

p_{Ao} = partial pressure of the absorbable constituent, atm.

P = total pressure, atm.

ρ = density of the liquid, lb. mols/ft.³

As Equations (6.13) and (6.14) hold for any point in the tower, it is evident that the operating lines pass through the points (X_2, Y_2) and (C_{AL_2}, p_{AG_2}) , where the subscript 2 refers to conditions at the top of the tower.

Equations (6.13) or (6.14) can be combined with a rate equation to give an expression which can be used to calculate the volume, or height, of tower required for a given absorption with a given ratio of liquid to

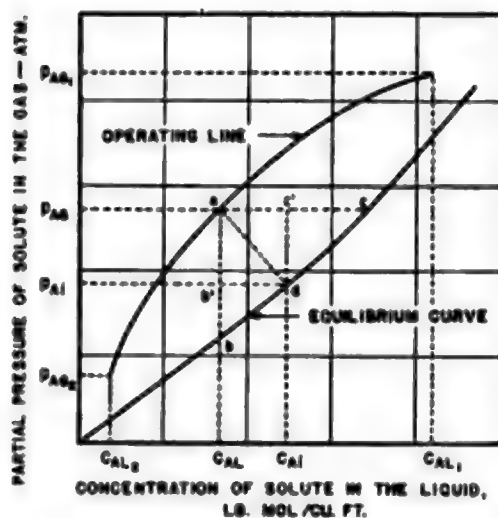


Fig. 2

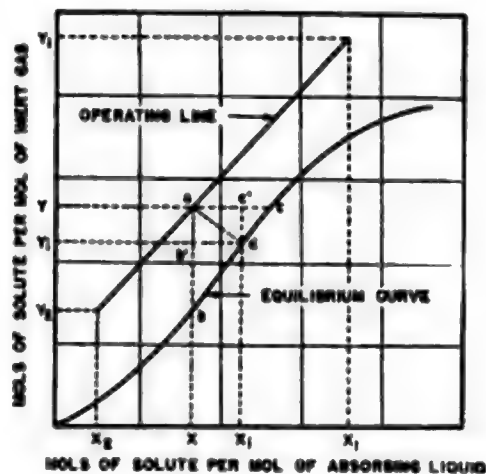


Fig. 3

Figure 2. Absorption Diagram for Tower Conditions in Terms of Partial Pressure and Concentration

Figure 3. Absorption Diagram for Tower Conditions in Terms of Mol Ratios

gas. However, the rate equation must first be written in differential form:

$$dN_A = k_G(p_{Ao} - p_{Ai})dA = k_G a(p_{Ao} - p_{Ai})dV \quad (6.15)$$

$$= k_L(C_{Ai} - C_{AL})dA = k_L a(C_{Ai} - C_{AL})dV \quad (6.16)$$

The subsequent procedure then depends upon whether the gas-film resistance is controlling, the liquid-film resistance is controlling, or both are important.

Gas-Film Resistance Controlling. Where the gas is highly soluble in the absorbing liquid, the resistance of the liquid film is frequently very small compared to the gas-film resistance and, for practical purposes, may be neglected. In such cases, the partial pressure and the

concentration at the gas-liquid interface— p_{Ai} and C_{Ai} —may be assumed to be the same as the corresponding values for the bulk of the liquid— p_{AL} and C_{AL} —and the rate equation may be written as

$$dN_A = k_G a (p_{AG} - p_{Ai}) dV = k_G a (p_{AG} - p_{AL}) dV \quad (6.17)$$

Furthermore, since

$$N_A = -L \int_{x_1}^{x_2} dx = -G \int_{Y_1}^{Y_2} dY \quad (6.18)$$

Equation (6.17) may be written as

$$k_G a (p_{AG} - p_{Ai}) dV = -L dX = -G dY \quad (6.19)$$

From Equations (6.13) and (6.14),

$$L(X_1 - X) = G \left(\frac{p_{AG_1}}{P - p_{AG_1}} - \frac{p_{AG}}{P - p_{AG}} \right) \quad (6.20)$$

which upon differentiation becomes

$$L dX = GP \frac{dp_{AG}}{(P - p_{AG})^2} \quad (6.21)$$

Equating $L dX$ from Equations (6.19) and (6.21), separating variables, and integrating:

$$-GP \frac{dp_{AG}}{(P - p_{AG})^2} = k_G a (p_{AG} - p_{Ai}) dV \quad (6.22)$$

and

$$\int_{V_1}^{V_2} dV = V = -GP \int_{p_{AG_1}}^{p_{AG_2}} \frac{dp_{AG}}{k_G a (P - p_{AG})^2 (p_{AG} - p_{Ai})} \quad (6.23)$$

or, if $k_G a$ may be considered constant

$$V = \frac{GP}{k_G a} \int_{p_{AG_1}}^{p_{AG_2}} \frac{dp_{AG}}{(P - p_{AG})^2 (p_{AG} - p_{Ai})} \quad (6.24)$$

If k_G is replaced by its equivalent value from Equation (6.5):

$$k_G a = \frac{D_G P a}{RT B_G p_{BM}} = \frac{b_G a}{B_G p_{BM}} \quad (6.25)$$

where

$$b_G = \frac{D_G P}{RT} \quad (6.26)$$

and

$$\begin{aligned}
 p_{BM} &= \frac{p_{Bi} - p_{Bg}}{\ln \frac{p_{Bi}}{p_{Bg}}} = \frac{(P - p_{Ai}) - (P - p_{Ag})}{\ln \frac{(P - p_{Ai})}{(P - p_{Ag})}} \\
 &= \frac{p_{Ag} - p_{Ai}}{\ln \frac{(P - p_{Ai})}{(P - p_{Ag})}} \quad (6.27)
 \end{aligned}$$

Equation (6.23) may also be written as

$$V = GP \int_{p_{Ag_2}}^{p_{Ag_1}} \frac{B_g}{b_g a} \frac{dp_{Ag}}{(P - p_{Ag})^2 \ln \frac{(P - p_{Ai})}{(P - p_{Ag})}} \quad (6.28)$$

In order to calculate the tower volume V by means of Equation (6.24), values of p_{Ag} are assumed, corresponding values of p_{Ai} —point b —are read from a diagram such as Figure 2, values of $1/[(P - p_{Ag})^2 (p_{Ag} - p_{Ai})]$ are calculated and plotted as ordinates against the corresponding values of p_{Ag} as abscissas, the area under the curve is determined graphically, and the value of the integral is multiplied by $GP/k_g a$. Equations (6.23) and (6.28) are solved in a similar manner.

An alternative procedure where $k_g a$ is constant is to make use of a mol ratio diagram, such as Figure 3. For, since

$$Y = \frac{p}{P - p} \quad (6.29)$$

or

$$p = \frac{PY}{1 + Y} \quad (6.30)$$

Equation (6.19) may be written as

$$-GdY = k_g a P \left(\frac{Y}{1 + Y} - \frac{Y_i}{1 + Y_i} \right) dV \quad (6.31)$$

$$= k_g a P \frac{Y - Y_i}{(1 + Y)(1 + Y_i)} dV \quad (6.32)$$

where

Y and Y_i = mols of absorbable constituent per mol of inert gas in the gas phase and at the gas-liquid interface, respectively.

Therefore,

$$\int_{r_1}^{r_2} dV = V = \frac{G}{k_o a P} \int_{r_1}^{r_2} \frac{(1 + Y)(1 + Y_i)}{Y - Y_i} dY \quad (6.33)$$

which can be solved graphically by assuming values of Y , reading off the corresponding values of Y_i —point b —, calculating values of $\frac{(1 + Y)(1 + Y_i)}{Y - Y_i}$, plotting them as ordinates against values of Y as abscissas, determining the area under the curve, and multiplying it by $\frac{G}{k_o a P}$.

In cases where the gas is very dilute, Y and Y_i are negligible compared to 1, and Equation (6.33) may be simplified to

$$V = \frac{G}{k_o a P} \int_{r_1}^{r_2} \frac{dY}{Y - Y_i} \quad (6.34)$$

Liquid-film Resistance Controlling. In the case of a very slightly soluble gas, the gas-film resistance is of negligible importance, and the rate of absorption is controlled by the liquid-film resistance. The equation to be used in this case may be derived as follows:

From Equations (6.13) and (6.14):

$$L(X_1 - X) = L \left(\frac{C_{AL_1}}{\rho_1 - C_{AL_1}} - \frac{C_{AL}}{\rho - C_{AL}} \right) \quad (6.35)$$

which upon differentiation becomes

$$LdX = L \frac{dC_{AL}}{(\rho - C_{AL})^2} \quad (6.36)$$

Equation (6.36) can be combined with Equation (6.16), remembering that $-LdX = dN$

$$-L \frac{dC_{AL}}{(\rho - C_{AL})^2} = k_L a (C_{Ai} - C_{AL}) dV \quad (6.37)$$

Therefore,

$$\int_{r_1}^{r_2} dV = V = L \int_{C_{AL_2}}^{C_{AL_1}} \frac{1}{k_L a (\rho - C_{AL})^2 (C_{Ai} - C_{AL})} dC_{AL} \quad (6.38)$$

or, if $k_o a$ is constant throughout the tower

$$V = \frac{L}{k_L a} \int_{C_{AL_2}}^{C_{AL_1}} \frac{dC_{AL}}{(\rho - C_{AL})^2 (C_{Ai} - C_{AL})} \quad (6.39)$$

This equation can be solved by graphical integration in the same manner as the equations for the conditions where the gas-film resistance is controlling. In this case, values of C_{AL} are assumed, values of C_{Ai} —point c —are read from a diagram such as Figure 2, and values of $1/[(\rho - C_{AL})^2(C_{Ai} - C_{AL})]$ are calculated and plotted against C_{AL} .

Equation (6.38) may be combined with Equation (6.6) and expressed in terms of the diffusivity and film thickness instead of $k_G a$:

$$V = \frac{L}{a} \int_{C_{AL_1}}^{C_{AL_2}} \frac{(B_L C_{GM}) dC_{AL}}{D_L(C_{AL} + C_{CL})(\rho - C_{AL})^2(C_{Ai} - C_{AL})} \quad (6.40)$$

If the mol-ratio diagram is used instead of the partial pressure-concentration diagram, and if X and X_i are the mols of absorbable constituent per mol of absorbing liquid in the liquid phase and at the gas-liquid interface, respectively, Equation (6.16) may be written as

$$dN_A = -LdX = k_L a \left[\frac{\rho_i X_i}{(1 + X_i)} - \frac{\rho_L X}{(1 + X)} \right] dV \quad (6.41)$$

since

$$X = \frac{C}{\rho - C} \quad (6.42)$$

or

$$C = \rho \left(\frac{X}{1 + X} \right) \quad (6.43)$$

If an average value of the density ρ_{av} is used, and if $k_L a$ may be assumed to be constant throughout the tower, Equation (6.41) becomes

$$V = \frac{L}{k_L a \rho_{av}} \int_{X_1}^{X_2} \frac{(1 + X_i)(1 + X) dX}{(X_i - X)} \quad (6.44)$$

The value of the integral can be obtained by plotting $(1 + X_i)(1 + X)/(X_i - X)$ against X and determining the area under the curve.

However, as the liquid-film resistance controls where the solubility of the gas is low and, under such conditions, X_i and X are small compared to 1, it is usually sufficiently accurate to simplify Equation (6.44) to

$$V = \frac{L}{k_L a \rho_{av}} \int_{X_1}^{X_2} \frac{dX}{(X_i - X)} \quad (6.45)$$

Where Both the Gas-Film and the Liquid-Film Resistances are Important. The equations previously given for conditions where either

the gas-film or the liquid-film resistance is controlling can be used with only slight modification where both resistances are important. In Figures 2 and 3, the driving force where the gas film is of controlling importance is represented by the line $a-b$, and the driving force where the liquid film is controlling is represented by the line $a-c$. If both resistances are important, the driving force is represented by some intermediate line such as $a-d$.

On the partial pressure-concentration diagram—Figure 2—line $a-d$ has a slope of $-k_L/k_G$, and on the mol ratio diagram—Figure 3—it has a slope of $-\frac{k_L(1+Y_{av})\rho_{av}}{k_G(1+X_{av})P}$ where X_{av} , Y_{av} , and ρ_{av} are the average values of (X and X_i), (Y and Y_i), and (ρ_L and ρ_i), respectively.

If lines of these slopes are constructed and projected to points b' and c' , values corresponding to the lengths $a-b'$ and $a-c'$ may be used in place of $(p_{AG} - p_{AL})$ or $(Y - Y_i)$, and $(C_{Ai} - C_{AL})$ or $(X_i - X)$ in the equations developed for conditions where either the gas-film or the liquid-film resistance is controlling.

Algebraic Method of Solution. If, over the range involved, Henry's law may be assumed to hold, and the quantities $(P - p_{AG})$ and $(\rho - C_{AL})$ do not change greatly, both the operating line and the equilibrium curve will be straight lines on either the partial pressure-concentration or the mol ratio diagram. Under such conditions, it is unnecessary to resort to graphical integration, for the absorption equation may be written as⁹

$$N_A = K_G a V \Delta p_M \quad (6.46)$$

or

$$N_A = \alpha K_G a V \Delta Y_M \quad (6.47)$$

where

$$\Delta p_M = \frac{\Delta p_1 - \Delta p_2}{2.303 \log_{10} \frac{\Delta p_1}{\Delta p_2}} = \frac{(p_{AG_1} - p_{AL_1}) - (p_{AG_2} - p_{AL_2})}{2.303 \log_{10} \frac{p_{AG_1} - p_{AL_1}}{p_{AG_2} - p_{AL_2}}} \quad (6.48)$$

$$\Delta Y_M = \frac{\Delta Y_1 - \Delta Y_2}{2.303 \log_{10} \frac{\Delta Y_1}{\Delta Y_2}} = \frac{(Y_1 - Y_{L_1}) - (Y_2 - Y_{L_2})}{2.303 \log_{10} \frac{Y_1 - Y_{L_1}}{Y_2 - Y_{L_2}}} \quad (6.49)$$

$$\alpha = \frac{P}{1 + Y_{av}} \quad (6.50)$$

In Equations (6.48) and (6.49), 1 and 2 refer to the terminal conditions in the tower, and

p_{AG} = partial pressure of the absorbable constituent in the gas phase, atm.

p_{AL} = partial pressure of the absorbable constituent in equilibrium with the liquid, atm.

Y = mols of absorbable constituent per mol of absorbing liquid.

Y_L = mols of absorbable constituent per mol of absorbing liquid in equilibrium with the gas.

Equations similar to (6.46) and (6.47) may also be developed in terms of $K_L a$ and ΔC or ΔX .

Theoretical Plate Concept. If absorption is carried out in a bubble-cap or other type of plate tower, the theoretical plate concept may be

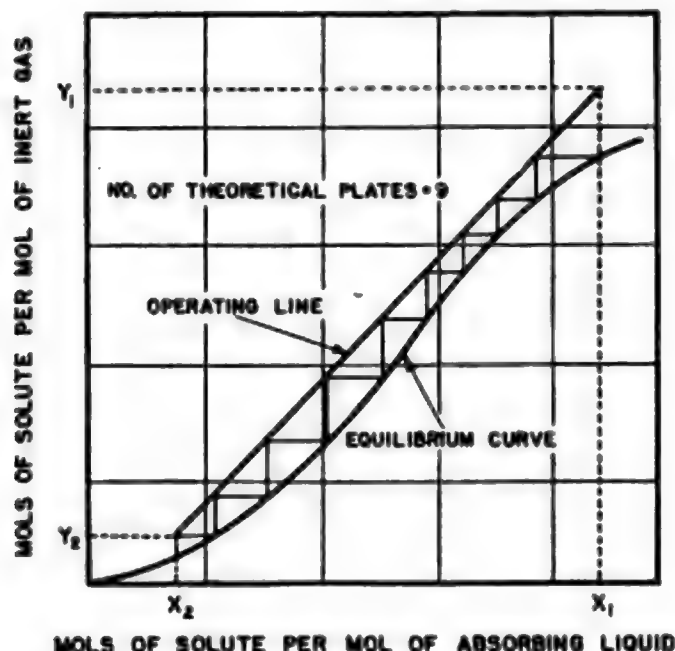


Figure 4. Graphical Determination of the Theoretical Number of Plates in an Absorption Tower

employed in the same manner as for distillation.³ This is illustrated in Figure 4, where the number of horizontal steps represents the number of theoretical plates in the tower.

The efficiency of an absorption tower, like the efficiency of a distillation column, may be expressed either in terms of the overall plate efficiency or in terms of the efficiencies of the individual plates. The overall efficiency is the ratio in per cent of the theoretical number of plates to the actual number, while the individual plate efficiency E_M has been defined by Murphree as¹⁰

$$E_M = \frac{Y_n - Y_{n+1}}{Y_n^* - Y_{n+1}} \times 100 \quad (6.51)$$

where

$(Y_n - Y_{n+1})$ = change in gas composition over the n th plate.

$(Y_n^* - Y_{n+1})$ = change in gas composition that would have occurred if equilibrium had been attained.

When the concentration of the solute in the gas and liquid is low and the equilibrium curve may be assumed to be a straight line, the number of theoretical plates n may be calculated algebraically, instead of graphically, by means of the equation¹⁷

$$\frac{Y_1 - Y_2}{Y_1 - mX_2} = \frac{\left(\frac{L}{mG}\right)^{n+1} - \left(\frac{L}{mG}\right)}{\left(\frac{L}{mG}\right)^{n+1} - 1} \quad (6.52)$$

where $m = Y/X$ is the slope of the equilibrium curve.

The theoretical plate concept may be applied to packed towers as well as to plate towers, and the capacity of a packed tower may be expressed in terms of the *H.E.T.P.*—the height equivalent to a theoretical plate.

Where the gas is very dilute and the equilibrium curve may be considered to be straight, the *H.E.T.P.* is related to K_0a by the equation

$$H.E.T.P. = \frac{G}{A'PK_0a} \quad (6.53)$$

where A' is the cross-sectional area of the tower in square feet. However, where the concentration of the solute is not low, the relationship between *H.E.T.P.* and K_0a is more complex.¹⁴

The Transfer Unit. Equation (6.21) may be rewritten and combined with Equations (6.15) and (6.25) to give:

$$-GP \frac{dp_{Ag}}{(P - p_{Ag})(P - p_{Ag})_{av}} = \frac{b_0 a A'}{B_0 p_{BM}} (p_{Ag} - p_{Ai}) dh \quad (6.54)$$

which when integrated becomes

$$\int_{p_{Ag_2}}^{p_{Ag_1}} \frac{dp_{Ag}}{(p_{Ag} - p_{Ai})} \frac{p_{BM}}{(P - p_{Ag})} = \frac{b_0 a A' (P - p_{Ag})_{av}}{B_0 PG} h \quad (6.55)$$

where

h = height of tower, ft.

$\frac{PG}{A'(P - p_{Ag})_{av}}$ = average total mols of solute plus inert gas per hr. per ft.² of tower cross-sectional area.

The left-hand side of Equation (6.55) has been defined by Chilton and Colburn⁶ as the number of *transfer units*. Therefore, the height of a transfer unit—*H.T.U.*—is:

$$H.T.U. = \frac{PG}{(b_{oa}/B_o)(A')(P - p_{Ao})_{av}} \quad (6.56)$$

or

$$H.T.U. = \frac{PG}{(k_o a p_{BM})(A')(P - p_{Ao})_{av}} \quad (6.57)$$

The *H.T.U.* is similar to the *H.E.T.P.*, but it is a more fundamental unit; it does not vary with the solute concentration and it varies only slightly with gas velocity.

If the operating and equilibrium lines are parallel, *H.T.U.* = *H.E.T.P.* Also, if the concentration of the solute in the gas phase is low, Equation (6.57) reduces to

$$H.T.U. = \frac{G}{A'Pk_o a} = \frac{G}{A'PK_o a} \quad (6.58)$$

which is identical with Equation (6.53) for the *H.E.T.P.* under the same conditions.

LABORATORY GAS-ABSORPTION TOWERS

Most laboratory towers are of the packed type; such towers are simple in design, they are easily constructed, and the data that they provide can be readily interpreted in terms of large scale, commercial equipment. The shell for this type of tower may be made of standard steel pipe, galvanized stove pipe, rolled and welded sheet steel, or sections of stoneware, depending upon the corrosive character of the materials being handled, the size of the tower, and the extent to which it is to be used. Unless the tower is constructed to obtain certain specific data, it should be flanged at both ends so that different packings may be easily interchanged.

Because laboratory towers are generally relatively small, the amount of absorption that takes place in the unpacked sections at the top and bottom of the tower may be very appreciable unless precautions are taken to minimize these *end effects*. This can be done by placing the liquor distributor immediately above the packing and placing the gas and liquid sampling tubes at the bottom of the tower immediately below the packing.

Many laboratory towers are too small in diameter, and lead to erroneous results because of the *channeling effect*. If the ratio of tower

diameter to packing size is less than about 8 to 1, there is a marked tendency for the liquid to concentrate near the walls and for the center to run nearly dry.^{4,8} Even in larger towers, channeling may be serious unless the liquor is uniformly distributed over the top of the packing. In any case, it is essential that the liquor sampler at the bottom of the tower be arranged so as to make it possible to obtain an average sample. It is also desirable to provide a means for obtaining samples of liquor at different points across the tower cross section so that any channeling will be made evident.

For fundamental investigations of mass transfer through films, the packed tower is not suitable because neither the interfacial area nor the conditions of turbulence are known. On the other hand, in a wetted-wall tower, the interfacial area is fixed and the conditions of flow of the gas stream can be regulated; for this reason, wetted-wall towers, although they are of no industrial importance, find a very definite place in the chemical engineering laboratory.

NOMENCLATURE

- a = effective absorbing area per unit of volume, $\text{ft.}^2/\text{ft.}^3$
- A = interfacial, or absorbing, area, ft.^2
- A' = cross sectional area, ft.^2
- A^* = constant.
- B = equivalent film thickness, ft.
- B^* = constant.
- C = concentration, lb. mols/ ft.^3
- d = tower diameter.
- D = diffusivity, $\text{ft.}^2/\text{hr.}$
- E_M = Murphree plate efficiency.
- G = gas rate, lb. mols of inert gas per hr.
- h = height of tower, ft.
- H = Henry's law constant.
- $H.E.T.P.$ = height equivalent to a theoretical plate.
- $H.T.U.$ = height of a transfer unit.
- k_g = gas film coefficient, lb. mols/(hr. \times $\text{ft.}^2 \times$ atm.).
- k_L = liquid film coefficient, lb. mols/(hr. \times $\text{ft.}^2 \times$ lb. mols/ ft.^3).
- K_g = overall absorption coefficient, lb. mols/(hr. \times $\text{ft.}^3 \times$ atm.).
- K_L = overall absorption coefficient, lb. mols/(hr. \times $\text{ft.}^2 \times$ lb. mols/ ft.^3).
- L = liquor rate, lb. mols of absorbing liquid per hr.
- m = slope of the equilibrium curve.
- M = molecular weight.

MF = mol fraction.

n = number of theoretical plates.

n' = constant.

N_A = rate of absorption, lb. mols/hr.

p = partial pressure, atm.

P = total pressure, atm.

P' = vapor pressure, atm.

R = gas constant, ft.³-atm./(lb. mol \times °R.).

T = absolute temperature, °R.

T' = absolute temperature, °K.

u = gas velocity.

V = volume of tower, ft.³

V' = molecular volume.

X = lb. mols of absorbable component per mol of absorbing liquid.

Y = lb. mols of absorbable component per mol of inert gas.

Y_n^* = lb. mols of absorbable constituent per mol of inert gas in equilibrium with the liquor on the n th plate.

$\alpha = P/(1 + Y_{av})$.

α' = constant.

μ = viscosity, lb./(ft. \times hr.).

ρ = density, lb./ft.³

Subscripts

1 = conditions at bottom of tower.

2 = conditions at top of tower.

A = soluble component.

av = average.

B = inert gas.

C = absorbing liquid.

G = gas.

i = interface.

L = liquid.

M = mean.

n = n th plate.

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A WETTED-WALL GAS-ABSORPTION SYSTEM

Designed and Constructed by

The Department of Chemical Engineering
Princeton University*

Description. This system, which was designed primarily for the study of the absorption of ammonia gas from ammonia-air mixtures using water as the absorbing liquid, is constructed almost entirely of standard pipe and fittings and a section of Pyrex tubing. The ammonia-air mixture passes through several feet of 2-inch pipe, which acts as a calming section, and then rises through the glass absorption tube where it encounters a descending film of water.

Materials and Cost. Figure 1 is a drawing of the assembled absorption system and gives the dimensions of the unit as constructed at *Princeton University*. About \$50 worth of materials were used in the construction of the equipment as illustrated; a bill of these materials is given in Table 1. However, there is nothing critical about the dimensions of the system, and they can be modified if desired.

Notes on Construction. 1. Glass absorbing section *M* is a 4-foot length of standard-wall Pyrex tubing flared at the bottom to a diameter of approximately 3 inches so as to fit over the inlet pipe and yet provide space for the liquor to flow out into liquor outlet glass *V*. In experiments with this apparatus, both 51-mm. and 48-mm. tubes are used; these tubes have internal diameters of 1.844 and 1.724 inches, respectively.

2. Water-inlet glass *R* and liquor-outlet glass *V* are sections of Pyrex, 6 inches in diameter. Each of these is drawn tight by stay bolts against two steel plates, which are covered with a resilient rubber packing cemented in place.

3. The liquor-discharge line is a length of rubber tubing *S* attached to a piece of glass tubing which, in turn, projects above the lower end of the absorption tube to provide a liquid seal, which permits the discharge of liquor without loss of gas. This tube is arranged so that it can be raised or lowered, for it is necessary to increase the liquor level as the pressure of the gas is increased.

4. The 2-inch line which carries the gas to the absorption tube is constructed in the form of a U to provide the proper calming section without excessive height of apparatus. With this arrangement, the

* Reported by J. C. Whitwell.

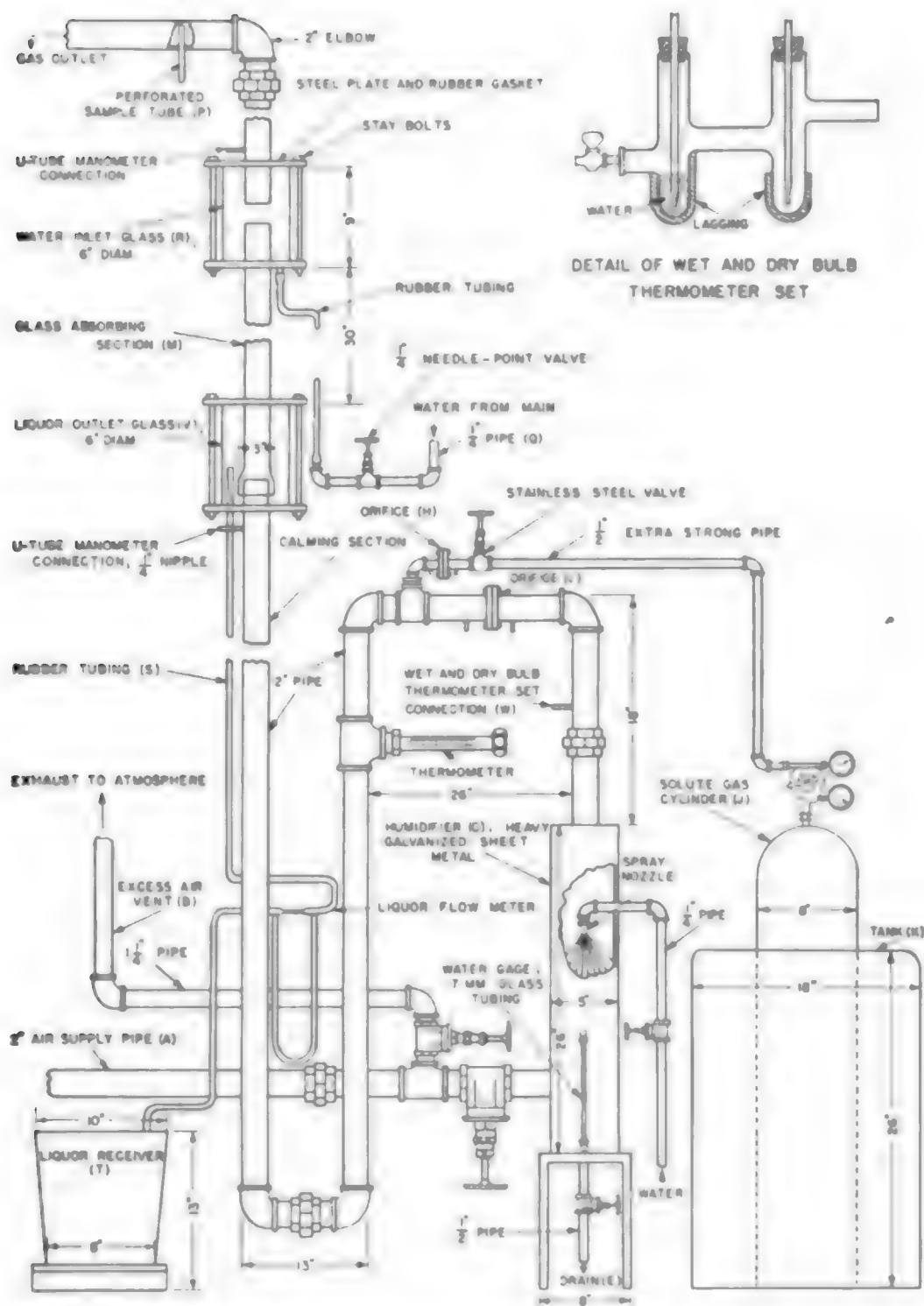


Figure 1. Wetted-Wall Gas-Absorption System—Princeton University

TABLE 1
Bill of Materials for Wetted-Wall Absorption System
 Princeton University

Quantity	Description	Size	Material
5 ft.	Extra strong pipe	$\frac{1}{2}$ in.	Black wrought steel
30 ft.	Std. pipe	2 in.	" " "
3 ft.	" "	$1\frac{1}{2}$ in.	" " "
5 ft.	" "	$\frac{1}{2}$ in.	" " "
10 ft.	" "	$\frac{1}{2}$ in.	" " "
1	Std. nipple	2 in. x 3 in. long	" " "
4	" "	2 in., short	" " "
2	" "	$1\frac{1}{2}$ in., close	" " "
1	" "	$\frac{1}{2}$ in. x 4 in. long	" " "
1	" "	$\frac{1}{2}$ in. x 6 in. "	" " "
1	" "	$\frac{1}{2}$ in. x 3 in. "	" " "
2	" "	$\frac{1}{2}$ in., short	" " "
7	" "	$\frac{1}{2}$ in. x 2 in. long	" " "
1	Std. tee	2 x 2 x $1\frac{1}{2}$ in.	Black malleable iron
2	" "	2 x 2 x 1 in.	" " "
5	Std. 90° elbow	2 in.	" " "
2	" " "	$1\frac{1}{2}$ in.	" " "
1	" " "	$\frac{1}{2}$ in.	" " "
6	" " "	$\frac{1}{2}$ in.	" " "
2	90° elbow, 300 lb.	$\frac{1}{2}$ in.	" " "
3	Ground-joint union	2 in.	" " "
1	Bushing	1 x $\frac{1}{2}$ in.	Cast iron
1	Std. flange union, gasket type	2 in.	" "
1	Std. flange union, gasket type	$\frac{1}{2}$ in.	" "
1	Std. gate valve	2 in.	Brass
1	" " "	$1\frac{1}{2}$ in.	"
1	" " "	$\frac{1}{2}$ in.	"
1	" " "	$\frac{1}{2}$ in.	"
1	Globe valve, 300 lb.	$\frac{1}{2}$ in.	Stainless steel
1	Needle-point valve	$\frac{1}{2}$ in.	Brass
1	Sheet	16 sq. ft. x 16-gage	Galv. steel
6 ft.	Angles	$\frac{3}{4}$ x $\frac{1}{2}$ x $\frac{1}{2}$ in.	Steel
4	Plate	9 in. O.D. x $\frac{1}{2}$ in. thick	"
8	Stay-bolts and nuts	$\frac{1}{2}$ in. -20 NC x 10 in. long	"
4 ft.	Tubing	51 mm. O.D.	Pyrex Glass
4 ft.	"	48 mm. O.D.	" "
2	Cylinder	6 in. diam. x 8 in. long	" "
6 ft.	Tubing	8 mm.	Glass
12 ft.	"	$\frac{1}{2}$ in. I.D.	Rubber
1	Spray nozzle	$\frac{1}{2}$ in.	Brass
4	Packing	9 in. O.D. x $\frac{1}{2}$ in. thick	Rubber

calming section, free from fittings which would disturb the flow, extends almost to the floor, while the thermometer well, orifice meter, and ammonia inlet are located in the other branch of the U.

Auxiliary Equipment. In performing an experiment with this absorption system, the following additional equipment is needed:

1. Wet gas meter.
2. Platform scales: 10-pound capacity.
3. Weighing bucket: 2-gallon capacity.
4. Air compressor and surge tank.
5. Cylinder of ammonia and regulating valves.

Operation. 1. Air is supplied by a Connersville blower of 88 cubic feet per minute maximum capacity operated by a 1-H.P. motor. To eliminate fluctuations in flow, a 50-gallon drum is inserted in the line between the blower and supply pipe *A*. The desired amount of air passes into humidifier *C*, while the excess is vented through pipe *B*. In the humidifier, the air becomes nearly saturated with water from the spray nozzle, and then passes through orifice *L* before it is mixed with the ammonia which enters through orifice *H*.

2. Excess water from the humidifier is removed through pipe *E*. To prevent loss of air through this pipe, a layer of water must be maintained in the bottom of the humidifier. This level is controlled by means of the valve in the bottom of the humidifier.

3. Ammonia enters the system from cylinder *J*, which is set in a large tank of water *K* to eliminate excessive cooling as the ammonia evaporates. After the ammonia meets the air stream, the mixed gases pass through the calming section and into glass absorbing section *M*. Unabsorbed gases leave through the gas outlet, which is provided with perforated sampling tube *P*.

4. Water from the main enters through line *Q* and passes into water-inlet glass *R*, from which it overflows into the top of glass absorbing section *M*. A $\frac{1}{4}$ -inch needle valve in the water line provides sufficiently close regulation of the rate of flow.

5. The liquor flows from the bottom of the absorption section into the liquor-outlet glass, from which it overflows into tube *S* and passes through the glass flow meter into weighing bucket *T*. The meter serves as a convenience in setting flow rates, but actual rate of discharge is measured by timing a given weight.

Determination of the Humidity. To determine the humidity of the air leaving the humidifier, the wet- and dry-bulb thermometer set, shown in the detail of Figure 1, is used. In operation, air is drawn through the set by means of a vacuum pump.

Analysis of the Gas. The concentration of the ammonia in the air-ammonia streams is determined by means of the apparatus shown in Figure 2. During an analysis, the gas stream is diverted through the absorption bottles and the wet gas meter; the ammonia is absorbed in standard acid, and the volume of ammonia-free air is then measured. The quantity of ammonia removed from the air is determined by back-titrating the acid solution with standard alkali.

Choice of the Gaseous Mixture. The ammonia-air-water system is very satisfactory for student work because it has the following advantages:

1. The equilibrium curve is substantially a straight line over the region involved.
2. As ammonia is a very soluble gas, the overall coefficient is practically equal to the gas-film coefficient.

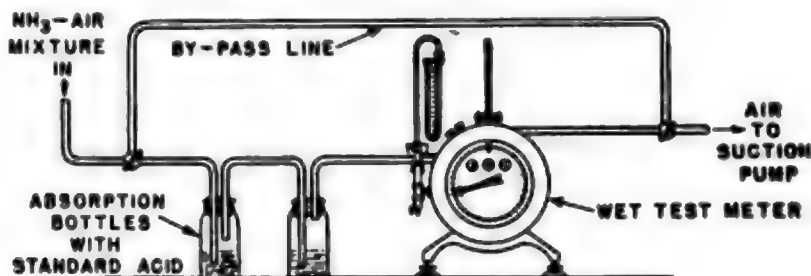


Figure 2. Apparatus for Determination of Ammonia in the Air-Ammonia Stream

3. Analysis of both gas and liquid can be carried out rapidly and accurately. In most cases, values of ammonia absorbed as obtained from analysis of the exit liquor agree within a few per cent with those obtained from analysis of the gas streams.

The *sulfur dioxide-air-water system* also gives good results. Except for analysis, the procedure is practically identical with that outlined for the ammonia-air-water system.

TYPICAL EXPERIMENT

Object. To determine the rate of absorption of ammonia gas in water under different conditions of concentration, liquor velocity, and gas velocity.

Procedure. 1. Start the flow of water and air through the absorption tube, adjusting their rates to approximately the desired values, and then open the valves in the ammonia line to give an air-ammonia mixture of approximately the desired concentration.

2. Attach the wet- and dry-bulb thermometer set to outlet *W*, turn on the water spray to the humidifier, and adjust the water rate until

it is sufficient to substantially saturate the air, as indicated by humidity determinations.

3. Record the following data at intervals of time until they indicate that conditions have become substantially constant, and then repeat the experiment under several different conditions:

- a. Velocities of the entering and exit air-ammonia streams.
- b. The rate of liquor flow.
- c. Analyses of the entering and exit gas and exit liquor.
- d. Temperature of the entering gas and the exit liquor.

4. For each run, calculate:

- a. The overall transfer coefficient, K_g .
- b. The Reynolds' number, Re .
- c. The height of a transfer unit, $H.T.U.$
- d. The values of $\frac{(D/B_g)}{(\mu/\rho b_g)^{0.44}}$
- e. The values of $\frac{H.T.U.}{D(\mu/\rho b_g)^{0.67}}$

5. Plot the following on log-log paper:

- a. K_g versus Re .
- b. Re versus $\frac{(D/B_g)}{(\mu/\rho b_g)^{0.44}}$ (Sherwood-Gilliland line).
- c. Re versus $\frac{H.T.U.}{D(\mu/\rho b_g)^{0.67}}$ (Chilton-Colburn line).

6. Compare the experimental results with those calculated from the Sherwood-Gilliland and the Chilton-Colburn equations.

Typical Data.* Experimental results for several runs are given in Table 2 and in Figures 3, 4 and 5.** The curve of Figure 3 is a plot of the experimentally determined values of the overall coefficient K_g versus Reynolds' number, Re . The curve of Figure 4, however, is a plot of the *Sherwood-Gilliland* equation¹

$$\frac{D}{B_g} = 0.023 \left(\frac{Du\rho}{\mu} \right)^{0.83} \left(\frac{\mu}{\rho D} \right)^{0.44} \quad (1)$$

although the points in this figure are experimentally determined values.

The dotted line of Figure 5 is a plot of the *Chilton-Colburn* equation^{1, 2}

$$H.T.U. = 10.9 D (\mu/\rho b_g)^{0.67} \left(\frac{Du\rho}{\mu} \right)^{0.1} \quad (2)$$

and the heavy line of this figure is the curve obtained experimentally.

* Data from Princeton thesis by G. L. Barnes and C. S. Elicker.

** The figures are based on more data than are given in Table 2.

TABLE 2
Experimental Results for the Absorption of Ammonia Gas in Water in a Wetted-Wall Absorption Column

Item No.	Symbol	Definition	Run No.							
			1	2	3	4	5	6	7	8
1	D	Tower diameter, ft.	0.1537	0.1537	0.1537	0.1537	0.1537	0.1537	0.1537	0.1445
2	V_1	Lb. mols air per hr.	5.30	1.04	0.961	0.995	0.015	4.89	4.74	3.04
3	n_1	NH_3 in entering gas, lb. mols/hr.	0.365	0.0736	0.0668	0.0308	0.0623	0.1514	0.329	0.0940
4	Y_1	Mol fraction NH_3 in entering gas.	0.0643	0.0661	0.0650	0.0300	0.0300	0.0300	0.0649	0.0300
5	t_{g1}	Temp. of entering gas, °C.	16.5	24.0	23.0	23.0	20.0	21.5	20.1	19.8
6	P_1	Press. of entering gas, mm. Hg.	770.8	765.1	760.6	760.3	760.4	744.8	744.8	765.4
7	P_{ag1}	Partial press. of NH_3 in entering gas, atm.	0.0652	0.0665	0.0651	0.0300	0.0300	0.0204	0.0636	0.0302
8	C_{ag1}	NH_3 in exit gas, per cent by volume.	4.78	3.91	3.85	1.70	1.923	2.065	4.76	2.225
9	P_2	Press. of exit gas, mm. Hg.	770.3	765.1	760.5	760.3	760.1	744.0	744.0	767.7
10	P_{ag2}	Partial press. of NH_3 in exit gas, atm.	0.0484	0.0393	0.0385	0.0170	0.01923	0.02021	0.0466	0.02245
11	w	Exit liquor rate, lb./hr.	106.9	107.6	92.0	88.6	89.5	91.1	90.0	93.6
12	C_{al1}	NH_3 in exit liquor, per cent by weight.	1.64	0.533	0.613	0.258	0.466	0.869	1.90	0.935
13	W_1	NH_3 absorbed, lb./hr. (from liquor titer).	1.753	0.573	0.664	0.228	0.417	0.792	1.710	0.596
14	X_1	Mol fraction NH_3 in exit liquor.	0.0173	0.00556	0.00648	0.00284	0.00491	0.00922	0.0205	0.0100
15	N_1	NH_3 absorbed, lb. mols/hr. (from liquor titer).	0.10293	0.03370	0.03315	0.01339	0.02449	0.04651	0.10040	0.03499
16	N_g	NH_3 absorbed, lb. mols/hr. (from gas analysis).	0.0990	0.0313	0.0263	0.0136	0.0228	0.0453	0.0921	0.0248
17	t_{l1}	Temp. exit liquor, °C.	20.5	18.2	20.0	19.4	18.5	20.0	23.0	20.1
18	P_{ol1}	Partial press. NH_3 in exit liquor, atm.	0.0146	0.0033	0.0046	0.0017	0.0032	0.0066	0.0184	0.0072
19	ΔP_m	Log-mean partial press. diff., atm.	0.0492	0.0501	0.0487	0.0222	0.0228	0.0215	0.0469	0.0227
20	K_g	Overall transfer coefficient, lb. mols/(hr. x ft. ² x atm.).	1.032	0.348	0.362	0.312	0.556	1.121	1.132	0.847
21	Re	Reynolds' number.	30,960	5,960	5,440	5,605	11,300	27,600	27,360	18,210
22	b_g	Diffusion coefficient, ft. ² /hr.	0.667	0.701	0.701	0.702	0.660	0.711	0.704	0.664
23	D/B_g	—	92.2	28.2	26.3	26.6	47.1	92.3	92.0	73.4
24	μ/ρ_b	—	0.844	0.841	0.826	0.838	0.837	0.836	0.839	0.833
25	$(D/B_g)/(\mu/\rho_b)^{0.45}$	—	99.3	30.4	31.6	28.8	51.0	101.0	99.3	78.4
26	$H.T.U.$	Height of a transfer unit, ft.	10.93	6.06	6.00	6.76	7.78	6.68	9.09	8.16
27	$H.T.U./D$	—	79.6	80.0	46.7	46.5	57.0	70.9	70.9	63.9

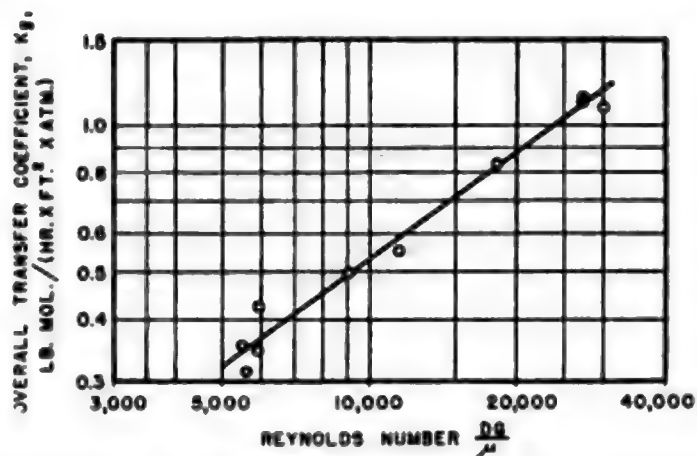


Figure 3. Experimental Overall Transfer Coefficients for Ammonia-Air-Water System

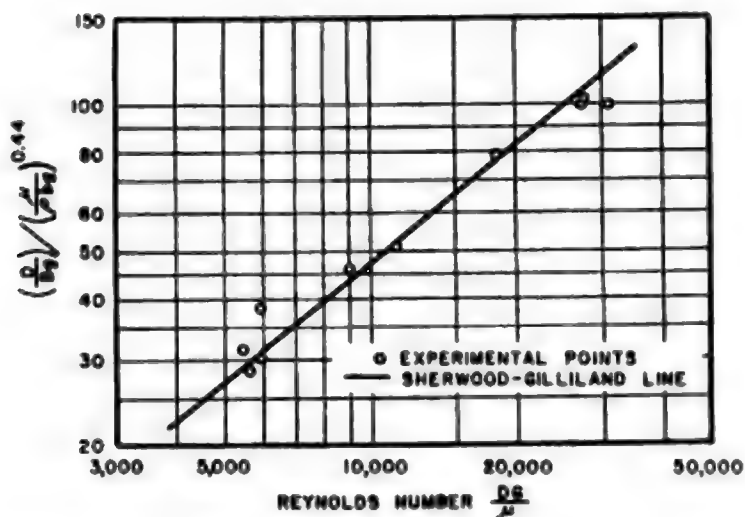


Figure 4. Comparison of Experimental Absorption Results for Ammonia-Air-Water System with Values Calculated from the Sherwood-Gilliland Equation

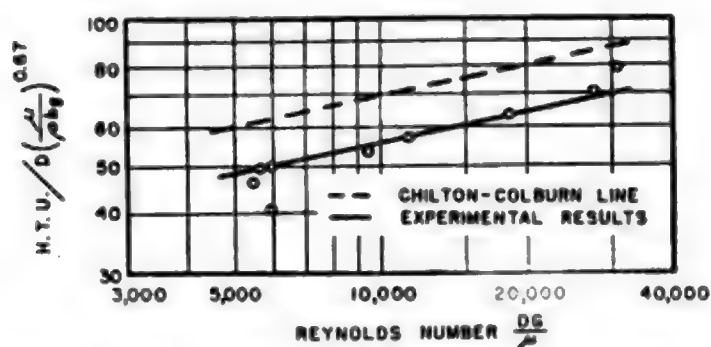


Figure 5. Comparison of Experimental Absorption Results for Ammonia-Air-Water System with Values Calculated from the Chilton-Colburn Equation

Sample Calculations. The treatment of the data are illustrated by the calculations for Run No. 1. The items in each case refer to values in Table 2.

Item 4: $Y_1 = \text{mol fraction of } NH_3 \text{ in entering gas}$

$$= \frac{n_1}{V_1 + n_1} = \frac{0.365}{5.30 + 0.365} = 0.0643$$

where

$V_1 = \text{lb. mols of inert gas per hr.} = \text{Item 2.}$

$n_1 = \text{lb. mols of } NH_3 \text{ in the entering gas per hr.} = \text{Item 3.}$

Item 7: $p_{a_{g1}} = \text{partial pressure of } NH_3 \text{ in entering gas, atm.}$

$$= Y_1 \times \frac{P'_1}{760} = 0.0643 \times \frac{770.8}{760} = 0.0652$$

where

$P'_1 = \text{pressure of entering gas, mm. Hg} = \text{Item 6.}$

Item 10: $p_{a_{g2}} = \text{partial pressure of } NH_3 \text{ in exit gas, atm.}$

$$= \frac{C_{a_{g2}}}{100} \times \frac{P'_2}{760} = \frac{4.78}{100} \times \frac{770.3}{760} = 0.0484$$

where

$C_{a_{g2}} = \text{per cent } NH_3 \text{ in exit gas} = \text{Item 8.}$

$P'_2 = \text{pressure of exit gas, mm. Hg} = \text{Item 9.}$

Item 13: $W_l = \text{lb. of } NH_3 \text{ absorbed per hr. (from liquor titer)}$

$$= w \times \frac{C_{a_{l1}}}{100} = 106.9 \times \frac{1.64}{100} = 1.753$$

where

$w = \text{exit liquor rate, lb. per hr.} = \text{Item 11.}$

$C_{a_{l1}} = \text{per cent } NH_3 \text{ in exit liquor} = \text{Item 12.}$

Item 14: $X_1 = \text{mol fraction of } NH_3 \text{ in exit liquor}$

$= 0.0173$ (This is calculated from concentrations using nomographic chart⁴).

Item 15: $N_l = \text{lb. mols of } NH_3 \text{ absorbed per hr. (from liquor titer)}$

$$= \frac{W_l}{\text{mol. weight } NH_3} = \frac{1.753}{17.03} = 0.10293$$

Item 16: $N_g =$ lb. mols of NH_3 absorbed per hr. (from gas analysis)

$$= n_1 - V_1 \times \frac{C_{as_1}}{100 - C_{as_1}}$$

$$= 0.365 - 5.30 \times \frac{4.78}{100 - 4.78} = 0.0990$$

Item 18: $p_{al_1} =$ partial pressure of NH_3 in exit liquor, atm.

$= 0.0145$ (This is calculated from concentrations using nomographic chart³).

Item 19: $\Delta p_m =$ log-mean partial pressure difference, atm.

$$= \frac{(p_{as_1} - p_{al_1}) - (p_{as_2} - p_{al_2})^*}{2.303 \log_{10} \frac{p_{as_1} - p_{al_1}}{p_{as_2} - p_{al_2}}}$$

$$= \frac{(0.0652 - 0.0145) - (0.0484 - 0)}{2.303 \log_{10} \frac{(0.0652 - 0.0145)}{(0.0484 - 0)}}$$

$$= 0.0492$$

Item 20: $K_g =$ overall transfer coefficient, lb. mols of NH_3 absorbed per hr. per ft.² per atm. partial pressure difference

$$= \frac{N_1}{(\text{wetted wall surface in ft.}^2) \times (\text{Item 19})}$$

$$= \frac{0.10293}{3.1416 \times 0.1537 \times 4 \times 0.0492} = 1.082$$

where

$N_1 = NH_3$ absorbed, lb. mols/hr. = Item 15.

wetted surface = $\pi \times$ diameter \times height

$$= 3.1416 \times 0.1537 \times 4$$

Item 21: $Re =$ Reynolds' number

$$= \frac{DG}{\mu} = \frac{0.1537 \times 8578}{0.0426} = 30,960$$

where

$D =$ diameter of column, ft. = 0.1537 = Item 1.

$G =$ average mass velocity of the gas, lb./(hr. \times ft.²)

$$= \frac{\text{lb. air/hr.} + \text{average lb. } NH_3/\text{hr.}}{\pi D^2/4}$$

* As the entering liquor was water in all of the runs, the partial pressure of ammonia in the entering liquor p_{al_1} was zero.

$$\begin{aligned}
 &= \frac{(29 \times V_1) + 17.03 (n_1 - \frac{1}{2} N_1)}{\pi D^2/4} \\
 &= \frac{(29 \times 5.30) + 17.03 (0.365 - \frac{1}{2} \times 0.10293)}{3.1416 \times (0.1537)^2/4} \\
 &= \frac{153.8 \times 5.34}{0.01855} = 8,578
 \end{aligned}$$

μ = average viscosity of the gas at 16.5°C., lb./ (ft. \times hr.)
 $= 2.42 \times$ (viscosity of the air in centipoises \times wt. fraction of the air + viscosity of the NH_3 in centipoises \times weight fraction of the NH_3 .)

$$\begin{aligned}
 &= 2.42 \left(0.0179 \times \frac{153.8}{159.14} + 0.010 \times \frac{5.34}{159.14} \right) \\
 &= 2.42 \times 0.0176 = 0.0426
 \end{aligned}$$

Item 22: b_g = diffusion coefficient, ft.²/hr.

This is calculated from the Gilliland equation⁴

$$\begin{aligned}
 b_g &= 0.01668 \times \frac{T^{3/2}}{P(V_a^{1/3} + V_b^{1/3})^2} \left(\frac{1}{M_a} + \frac{1}{M_b} \right)^{1/2} \\
 &= 0.01668 \times \frac{(289.6)^{3/2}}{1.014 (26.7^{1/3} + 29.9^{1/3})^2} \left(\frac{1}{17.03} + \frac{1}{29} \right)^{1/2} \\
 &= 0.667
 \end{aligned}$$

where

$$\begin{aligned}
 T &= \text{temperature of the gas } ^\circ\text{K.}, = t_{g1} + 273.1 \\
 &= 16.5 + 273.1 = 289.6
 \end{aligned}$$

$$\begin{aligned}
 P &= \text{av. pressure of the gas, atm.} = \frac{\frac{1}{2}(P'_1 + P'_2)}{760} \\
 &= \frac{\frac{1}{2}(770.8 + 770.3)}{760} = 1.014
 \end{aligned}$$

V_a = molecular volume of NH_3 = 26.7

V_b = molecular volume of air = 29.9

M_a = molecular weight of NH_3 = 17.03

M_b = molecular weight of air = 29

$$\begin{aligned}
 \text{Item 23: } D/B_g &= \frac{K_g RT(p_b)_m D}{b_g P} \\
 &= \frac{1.082 \times 0.73 \times 521.7 \times 0.981 \times 0.1537}{0.667 \times 1.014} = 92.2
 \end{aligned}$$

where

$$K_g = \text{overall transfer coefficient, lb. mol}/(\text{hr.} \times \text{ft.}^2 \times \text{atm.}) \\ = 1.082$$

$$R = \text{gas constant, (atm.} \times \text{ft.}^3)/(\text{lb. mol} \times ^\circ\text{R.}) = 0.73$$

$$T = \text{temp. of the gas, } ^\circ\text{R.} = (16.5 \times 9/5) + 32 + 460 \\ = 521.7$$

$$(p_b)_m = \text{mean partial pressure of the inert gas in the film,} \\ \text{atm.}^*$$

$$= \frac{1}{4} \left[\left(\frac{P'_2}{760} - p_{a_{O_2}} \right) + \left(\frac{P'_3}{760} - p_{a_{N_2}} \right) \right. \\ \left. + \left(\frac{P'_1}{760} - p_{a_{O_2}} \right) + \left(\frac{P'_1}{760} - p_{a_{N_2}} \right) \right] \\ = \frac{1}{4} \left[\left(\frac{770.3}{760} - 0.0484 \right) + \left(\frac{770.3}{760} - 0 \right) \right. \\ \left. + \left(\frac{770.8}{760} - 0.0652 \right) + \left(\frac{770.8}{760} - 0.0145 \right) \right] \\ = 0.981$$

$$D = \text{diameter of tower, ft.} = 0.1537 = \text{Item 1.}$$

$$b_g = \text{diffusion coefficient, ft.}^2/\text{hr.} = 0.667$$

$$P = \text{average total pressure of the gas, atm.} = 1.014$$

$$\text{Item 24: } \mu/\rho b_g = \frac{\mu R T}{b_g M P} \\ = \frac{0.0426 \times 0.73 \times 521.7}{0.667 \times 28.39 \times 1.014} \\ = 0.844$$

where

$$\mu = \text{mean viscosity of the gas, lb.}/(\text{ft.} \times \text{hr.}) = 0.0426$$

$$R = \text{gas constant, (atm.} \times \text{ft.}^3)/(\text{lb. mol} \times ^\circ\text{R.}) = 0.73$$

$$T = \text{temp. of the gas, } ^\circ\text{R.} = 521.7$$

$$b_g = \text{diffusion coefficient, ft.}^2/\text{hr.} = 0.667 = \text{Item 22.}$$

$$M = \text{average molecular weight of the gas}$$

$$= \frac{\text{average weight of gas per hr., lb.}}{\text{average lb. mols of gas per hr.}}$$

$$= \frac{\text{lb. air/hr.} + \text{av. lb. NH}_3/\text{hr.}}{\text{mols air/hr.} + \text{av. mols NH}_3/\text{hr.}}$$

* In this case the arithmetic mean is sufficiently accurate.

$$\begin{aligned}
 &= \frac{(29 \times V_1) + 17.03(n_1 - \frac{1}{2}N_t)}{\text{Item 2} + \text{Item 3} - \frac{1}{2} \text{Item 15}} \\
 &= \frac{(29 \times 5.30) + 17.03 (0.365 - \frac{1}{2} \times 0.10293)}{5.30 + 0.365 - \frac{1}{2} \times 0.10293} \\
 &= 28.39
 \end{aligned}$$

$$P = \text{average pressure of the gas, atm.} = 1.014$$

$$\begin{aligned}
 \text{Item 25: } \frac{D/B_g}{(\mu/\rho b_g)^{0.44}} \\
 &= \frac{\text{Item 23}}{(\text{Item 24})^{0.44}} \\
 &= \frac{92.2}{(0.844)^{0.44}} = 99.3
 \end{aligned}$$

$$\begin{aligned}
 \text{Item 26: } H.T.U. &= \frac{G}{K_g a (p_b)_m M} \\
 &= \frac{8,578}{1.082 \times 26.0 \times 0.981 \times 28.39} = 10.93
 \end{aligned}$$

where

$$G = \text{av. mass velocity of the gas, lb./}(\text{hr.} \times \text{ft.}^2) = 8,578$$

$$M = \text{av. molecular wt. of the gas} = 28.39$$

$$K_g = \text{overall transfer coefficient, lb. mols}/(\text{hr.} \times \text{ft.}^2 \times \text{atm.})$$

$$= \text{Item 20} = 1.082$$

$$a = \text{ft.}^2 \text{ of interfacial area per ft.}^3 \text{ of tower volume}$$

$$= \frac{\pi D \times \text{length}}{\pi D^2/4 \times \text{length}} = \frac{4}{D} = \frac{4}{0.1537} = 26.0$$

$$(p_b)_m = \text{mean partial pressure of the inert gas in the film, atm.} = 0.981$$

$$\begin{aligned}
 \text{Item 27: } \frac{H.T.U.}{D(\mu/\rho b_g)^{0.67}} \\
 &= \frac{\text{Item 26}}{\text{Item 1} \times (\text{Item 24})^{0.67}} = \frac{10.93}{0.1537 \times (0.844)^{0.67}} = 79.6
 \end{aligned}$$

Discussion of Results. The results of this experiment, as plotted in Figures 3, 4, and 5, show that reasonable accuracy can be obtained. In Figure 3, the experimental points fall very close to a straight line, and

in Figure 4 the experimental values agree reasonably well with those calculated from the Sherwood-Gilliland equation. However, there is some discrepancy between the plot of the Chilton-Colburn equation and the curve obtained experimentally; both curves have approximately the same slope, but the experimentally determined constant is 8.8 instead of 10.9. It is possible that some of the discrepancy in the results may have been due to entrance effects or rippling of the surface.

This absorption apparatus can be used over a fairly wide range of gas and liquid velocities, but the results indicate that best results are obtained between Reynolds' numbers of 10,000 to 30,000 and with water-flow rates of approximately 50 to 100 pounds per hour.

NOMENCLATURE

- a = ft.² of interfacial area per ft.³ of tower volume.
- b_g = diffusion coefficient, ft.²/hr.
- B_g = equivalent film thickness, ft.
- $C_{a,g}$ = ammonia in exit gas, per cent by volume.
- $C_{a,l}$ = ammonia in exit liquor, per cent by weight.
- D = tower diameter ft.
- G = average mass velocity of the gas, lb./(hr. \times ft.²).
- $H.T.U.$ = height of a transfer unit, ft.
- K_g = overall transfer coefficient, lb. mols/(hr. \times ft.² \times atm.).
- M = average molecular weight of gas.
- M_a = molecular weight of ammonia.
- M_b = molecular weight of air.
- n = ammonia in gas stream, lb. mols/hr.
- N_l = ammonia absorbed, lb. mols/hr. (from liquor titer).
- N_g = ammonia absorbed, lb. mols/hr. (from gas analysis).
- $p_{a,g}$ = partial pressure of ammonia in gas stream, atm.
- $(p_b)_m$ = mean partial pressure of inert gas in film, atm.
- $p_{a,l}$ = partial pressure of ammonia in liquor stream, atm.
- Δp_m = log mean partial pressure difference, atm.
- P' = pressure of the gas, mm. Hg.
- P = average pressure of the gas, atm.
- R = gas constant, (atm. \times ft.³)/(lb. mol \times °R.).
- t_g = temperature of the gas, °C.
- t_l = temperature of the liquid, °C.
- T = temperature of the gas °R.
- V_1 = lb. mols of air per hour.
- V_a = molecular volume of ammonia.
- V_b = molecular volume of air.

w = exit liquor rate, lb./hr.

W_1 = ammonia absorbed, lb./hr. (from liquor titer).

X = mol fraction of ammonia in liquor.

Y = mol fraction of ammonia in gas.

μ = viscosity, lb./(ft. \times hr.).

ρ = density of the gas, lb./ft.³

Subscripts

1 = conditions at bottom of tower.

2 = conditions at top of tower.

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A GAS-LIQUID ABSORPTION SYSTEM

Designed and Constructed by
The School of Chemical Engineering
Cornell University*

Description. This absorption system consists of: (1) a packed tower, approximately $7\frac{1}{2}$ feet high and 1 foot in internal diameter, in which liquid and gas are passed countercurrently; (2) a storage reservoir for the absorbing liquid; and (3) the necessary circulation systems for both liquid and gas. The liquid enters the top of the tower and passes into a basket with a perforated bottom which distributes it over the top of the packing. The liquid drops from the bottom of the packing into an unpacked space at the bottom of the tower, and overflows into the storage reservoir from where it is returned to the tower. The tower has a wetted packing area of 51 square feet and a total absorbing area, including the inner surface of the tower, of 64 square feet.

The assembly and details of construction of this absorption system are shown in Figure 1, and the materials used in its construction are listed in Table 1.

Notes on Construction. *Tower and Reservoir.* Tower *T* and liquor reservoir *R* are of all-welded construction; they were fabricated to order from sheet steel.

Outlet-liquor Sampler. Liquor sampler *S*₁ extends completely across the tower to permit the collection of an average sample of liquor. This sampler is made by closing one end of a length of $\frac{1}{2}$ -inch pipe and cutting a slot along the upper side.

Orifice Meters. The orifice meters *O*₁, *O*₂, and *O*₃ have orifices of $\frac{1}{2}$ -inch, $\frac{1}{4}$ -inch, and $\frac{3}{32}$ -inch, respectively. To provide sufficient difference between the velocity through the pipe and the velocity through the orifice, the lines are enlarged for a few feet on each side of the orifice. The orifice plates are made from sheet brass, about $\frac{3}{16}$ -inch thick; they are held in place between standard cast-iron flanges.

The manometer across orifice *O*₁, in the liquor line, contains mercury under a supernatant column of the liquid; the manometer across orifice *O*₂, in the outlet gas line, is a differential manometer containing carbon tetrachloride under water; and orifice meter *O*₃, in the solute gas line, is connected to two manometers: one containing carbon tetrachloride and the other mercury, for low and high rates of gas flow, respectively.

* Reported by C. C. Winding and F. H. Rhodes.

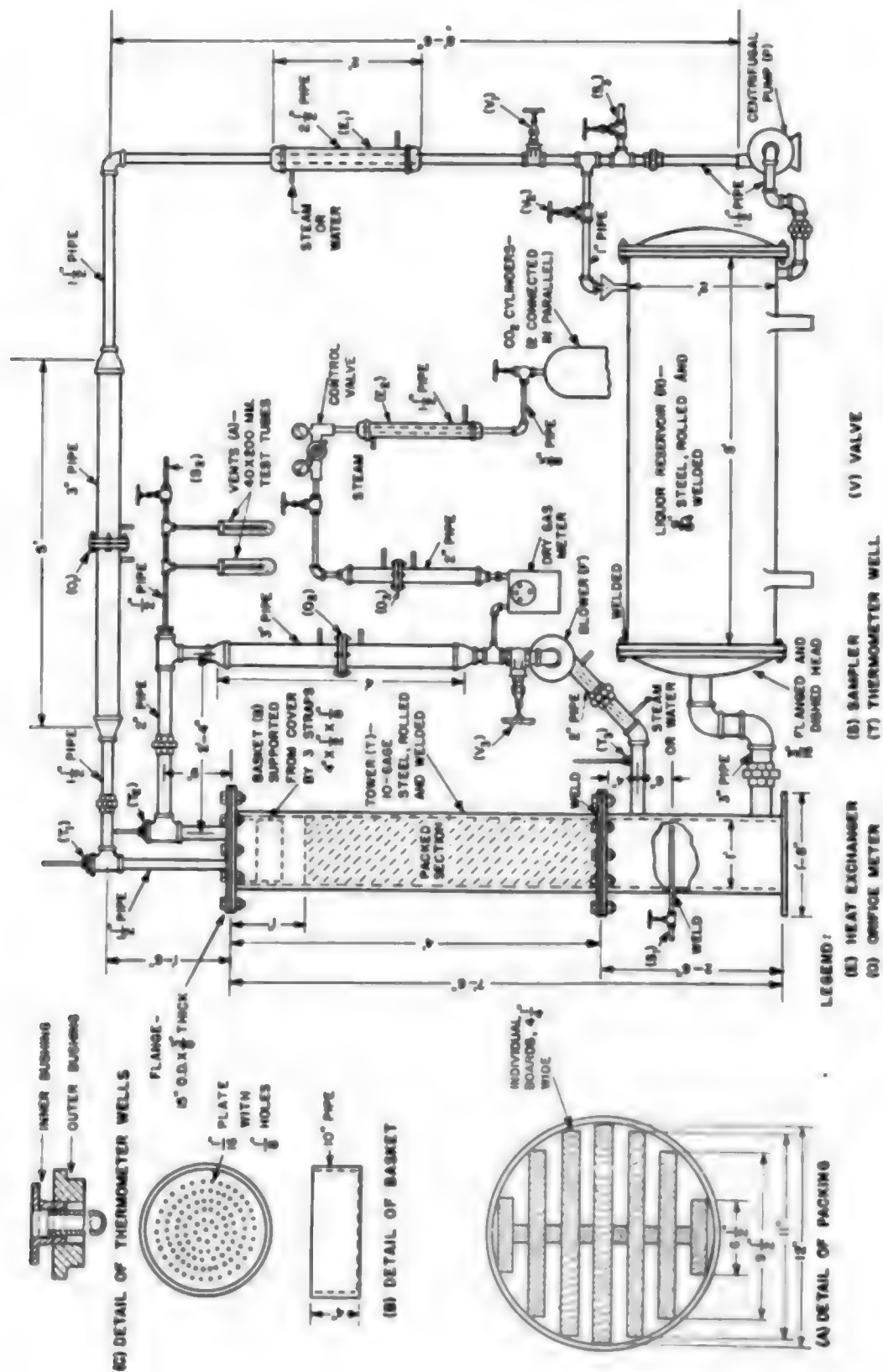


Figure 1. Gas-Liquid Absorption Column—Cornell University

TABLE 1
Bill of Materials for Gas-Liquid Absorption System
 Cornell University

Quantity	Description	Size	Material
4 in.	Std. pipe	10 in.	Black wrought steel
12 ft.	" "	3 in.	" " "
2 ft.	" "	2½ in.	" " "
3½ ft.	" "	2 in.	" " "
12½ ft.	" "	1½ in.	" " "
1½ ft.	" "	½ in.	" " "
3 ft.	" "	¾ in.	" " "
2 ft.	" "	½ in.	" " "
2	Std. nipple	3 in. x 4 in. long	" " "
1	" "	2 in. x 8 in. "	" " "
2	" "	2 in. x 7 in. "	" " "
1	" "	2 in. x 4 in. "	" " "
2	" "	2 in., short	" " "
1	" "	1½ in. x 9 in. long	" " "
3	" "	1½ in. x 6 in. "	" " "
2	" "	1½ in. x 4 in. "	" " "
4	" "	1½ in., short	" " "
1	" "	1 in. x 12 in. long	" " "
1	" "	1 in. x 6 in. "	" " "
2	" "	1 in. short	" " "
2	" "	½ in. x 6 in. long	" " "
1	" "	½ in. x 5 in. "	" " "
1	" "	½ in. x 4 in. "	" " "
2	" "	½ in. x 3 in. "	" " "
1	" "	¾ in. x 8 in. "	" " "
2	" "	¾ in. x 6 in. "	" " "
1	" "	¾ in. x 3 in. "	" " "
6	" "	½ in. x 4 in. "	" " "
4	" "	½ in. x 3 in. "	" " "
2	Std. tee	2 in.	Black malleable iron
1	" "	1½ in.	" " "
2	" "	1½ x 1½ x 1 in.	" " "
2	" "	½ in.	" " "
2	Std. 90° elbow	3 in.	" " "
4	" " "	1½ in.	" " "
1	" " "	1 in.	" " "
2	" " "	½ in.	" " "
2	" " "	¾ in.	" " "
1	Std. 45° elbow	2 in.	" " "
1	Ground-joint union	3 in.	" " "
2	" " "	2 in.	" " "
3	" " "	1½ in.	" " "

TABLE 1—*Concluded*

Quantity	Description	Size	Material
2	Bushing	2 x $\frac{1}{2}$ in.	Cast iron
1	"	2 x $\frac{1}{2}$ in.	" "
1	"	1 $\frac{1}{2}$ x $\frac{1}{2}$ in.	" "
3	"	$\frac{1}{2}$ x $\frac{1}{2}$ in.	Black malleable iron
2	Std. reducer	3 x 2 in.	" " "
2	" "	3 x 1 $\frac{1}{2}$ in.	" " "
2	" "	2 x $\frac{1}{2}$ in.	" " "
2	Std. cap	2 $\frac{1}{2}$ in.	" " "
2	" "	1 $\frac{1}{2}$ in.	" " "
1	Gate valve, 125 lb.	2 in.	Brass
1	" " " "	1 $\frac{1}{2}$ in.	"
2	" " " "	1 in.	"
1	" " " "	$\frac{1}{2}$ in.	"
1	" " " "	$\frac{1}{4}$ in.	"
1	Absorption tower	12 in. I.D. x 7 $\frac{1}{2}$ ft. high	Steel
1	Liquor storage reservoir	2 ft. diam. x 5 ft. long	"
1 sheet	Gasket material	4 ft. x 4 ft. x $\frac{1}{8}$ in. thick	Rubber
66 ft.	Lumber	1 x 4 $\frac{1}{2}$ in., dressed	Pine
1	Centrifugal pump and motor*	1 $\frac{1}{2}$ in.	—
1	Blower and motor, Cadillac Model D**	2 in.	—
3	Manometer	8 mm. tubing	Glass
4	Thermometer	-10 to 110°C.	Glass-mercury
2	Std. flange union; gasket type	3 in.	Cast iron
1	Std. flange union; gasket type	2 in.	" "
1	Round plate	6 $\frac{1}{2}$ in. O.D. x $\frac{3}{8}$ in. thick, with $\frac{1}{2}$ in. opening	Brass
1	" "	6 $\frac{1}{2}$ in. O.D. x $\frac{3}{8}$ in. thick, with $\frac{1}{2}$ in. opening	"
1	" "	5 $\frac{1}{2}$ in. O.D. x $\frac{3}{8}$ in. thick, with $\frac{1}{2}$ in. opening	"

* Mfgd. by Goulds Pumps, Inc., Seneca Falls, New York.

** Mfgd. by Clements Manufacturing Co., Chicago, Illinois.

Vents. The two vents *A* are large test tubes which provide water seals to control the pressure in the gas outlet line at atmospheric ± 1

inch of water; if the pressure becomes too great, gas escapes from the system, and if it becomes too low, air is drawn in.

Packing. The packing is made of a number of wooden grids, constructed as shown in Detail A. Alternate grids are assembled in the tower with their axes at right angles to each other.

Comments on Design. *Tower.* Sections of standard 12-inch pipe and screwed flanges may be used in place of the welded construction; however, this has the disadvantage of materially increasing the weight.

Liquor Reservoir. A steel drum or any other inexpensive tank may be used in place of the more costly fabricated reservoir.

Packing. Other types of packing may be used in place of the wood grid. The disadvantage of a wood grid is the long time lag in attaining equilibrium due to the slow absorption and release of the gases by the wood itself. If reliable data are to be obtained, as much as 24 hours may be required for conditions to become sufficiently constant.*

Cost. The cost of this absorption system as constructed at *Cornell University* is approximately \$325. This includes a tower and liquor reservoir of welded construction; a centrifugal pump and a blower with their respective motors; and the pipe, fittings, and miscellaneous materials necessary to complete the assembly. The tower and the liquor reservoir cost \$71 and \$53, respectively; and the pump, blower, and motors cost approximately \$100.

If the tower is constructed of standard pipe and flanges and if an oil drum or other inexpensive tank is substituted for the fabricated reservoir, the cost of materials can be reduced to approximately \$250.

Operation. The system is operated by circulating the absorbing liquor over the packing in the tower by means of pump *P* and passing the gas countercurrently by means of blower *F*.

The gas to be absorbed is introduced continually into the main gas stream at the suction side of the blower at a rate sufficient to maintain a constant concentration in the gaseous mixture entering the tower.

The temperature and the rate of flow of both liquor and gas are carefully controlled by means of heat exchangers, orifice meters, and valves in the respective lines. Samples of the gas leaving the column and samples of the entering and exit liquors are withdrawn for analysis periodically during a run.

Auxiliary Equipment. Although this apparatus can be used with a number of different gases and liquids, the carbon dioxide-caustic soda system is convenient for student operation. This system requires:

* Hougen, O. A., private communication to authors.

1. Equipment for delivering fresh, metered carbon dioxide; this includes:

- a. Two cylinders of carbon dioxide connected in parallel so that one cylinder can be replaced while the other is in service.
- b. Gas regulating and control valves.
- c. A dry gas meter provided with a thermometer and an open-end manometer containing carbon tetrachloride under water.

2. Equipment for sampling and analyzing the gas and liquor.

Sampling and Analysis. Gas Mixture. A Zeiss gas interferometer is used at Cornell University, although other methods of analysis may be used. The interferometer, which can be used only with gas mixtures containing a single variable component, provides a very rapid and accurate method for analyzing low concentrations of carbon dioxide in air, as used in this experiment. Duplicate analyses of mixtures of 1 to 2 per cent carbon dioxide in air can be made in 1 to 2 minutes with an accuracy of ± 0.03 per cent. With this rapid analysis, a complete run can be made in 6 to 10 minutes. A modified Hemple burette can also be used for the gas analysis, but this requires from 10 to 20 minutes for each sample.

The gas collecting equipment that is used with the interferometer consists of a glass reservoir of 200 ml. capacity provided at its upper end with a two-way stop cock and at its lower end with a one-way stop cock, which is attached to a section of rubber tubing that leads to a mercury-leveling bulb. One arm of the two-way stop cock leads to the chamber of the gas interferometer, and the other arm connects to sampling outlet S_2 through a U-tube containing glass beads wetted with concentrated sulfuric acid. One mm. capillary tubing is used for the connecting line.

In making an analysis with the Zeiss interferometer, the procedure is as follows:

1. The mercury-leveling bulb is lowered, the two-way stop cock is turned to connect the glass reservoir to sampling outlet S_2 , and a sample of the gas mixture is drawn through the U-tube and into the reservoir.
2. The stop cock is then turned so that the reservoir is connected to the chamber of the interferometer, and the leveling bulb is raised to force the gas through the instrument.
3. The gas is withdrawn from the interferometer and discarded, and a fresh sample is drawn into the apparatus; this is repeated again to thoroughly rinse the system.

4. A sample for analysis is then forced into one of the two halves of the measuring chamber of the interferometer, and air, free from moisture and carbon dioxide, is introduced under atmospheric pressure into the other half of the chamber. The air is freed of water and carbon dioxide by passing it through a wash bottle containing concentrated sulfuric acid and then through a soda-lime absorption tower.

5. The reading of the interferometer is taken, and the percentage of carbon dioxide in the dry gas is determined from the calibration chart for the instrument.

Absorbing Liquor. One pint samples of the absorbing liquor are taken at the beginning and end of each run and at one or two intervals during the run. The density of each sample at 25°C. is measured with a hydrometer, and each sample is then analyzed in the following manner:

1. A 50 ml. portion is pipetted into a 500 ml. volumetric flask, which is then filled to the mark with carbon dioxide-free distilled water. The mixture is thoroughly shaken, and several 50 ml. portions are pipetted into separate 250 ml. Erlenmeyer flasks.

2. A drop of phenolphthalein solution is added to two or more of the 50 ml. portions and they are titrated with standard 0.5 N hydrochloric acid until the pink color just disappears; a drop of methyl orange solution is then added and the titration is continued to the methyl orange end point. Titration to the phenolphthalein end point neutralizes all of the hydroxide and converts the carbonate to the bicarbonate; titration to the methyl orange end point neutralizes the resulting bicarbonate.

3. A slight excess of 5 per cent barium chloride solution is added to two other 50 ml. portions to react with the carbonate present. The quantity of barium chloride solution required is calculated from the difference between the volumes of acid required in the previous titrations to reach the phenolphthalein and the methyl orange end points. The flasks are thoroughly shaken, two drops of phenolphthalein solution are added to each, and the contents are titrated with standard hydrochloric acid solution.

This titration, which gives only the quantity of hydroxide present in the sample, is performed because it is more accurate than the double titration method.

4. The quantity of carbonate is then calculated from the total acid required in the double titration and the quantity used to neutralize the hydroxide.

TYPICAL EXPERIMENT

Object. To study the influence of the gas and liquor rates on the rate of absorption of carbon dioxide by a caustic soda solution.

Procedure. *Preparation of the Caustic Soda Solution.* Prepare, in storage reservoir *R*, a solution containing approximately 100 pounds of commercial flake caustic soda (76% Na_2O) and 900 pounds of water. This solution is most easily prepared by circulating liquor over the caustic while it is held in a large funnel, provided with a coarse screen bottom, placed in the inlet of the reservoir. During this operation valve V_1 is closed.

After complete solution of the caustic, close valve V_2 and open valve V_1 . Take a sample from S_1 and analyze it for the total per cent Na_2O . Adjust the concentration of the caustic to the desired value by adding water or flake caustic soda as needed.

Circulation of the Liquor. Circulate the prepared solution through the tower, adjust valve V_1 to give the desired rate of flow, and pass cold water or steam through heat exchanger E_1 to maintain the temperature of the liquor entering the tower at approximately 25°C .

Circulation of the Gas. Start the blower, adjust the rate of air flow by means of valve V_3 , and then introduce carbon dioxide from the side stream at a rate sufficient to maintain its concentration at about 3 per cent in the air-carbon dioxide mixture entering the tower. Pass steam through heat exchanger E_2 so that the temperature of the carbon dioxide, as measured by a thermometer at the dry gas meter, is approximately the same as the liquor temperature.

Data. When conditions become constant, record the following data:

1. Readings of the orifice manometer in the outlet gas line.
2. Readings of the orifice manometer in the liquor line.
3. Readings of the orifice manometer in the carbon dioxide line.
4. Temperature of the inlet liquor.
5. Temperature of the outlet liquor.
6. Temperature of the entering gas.
7. Temperature of the exit gas.
8. Temperature of the carbon dioxide at the dry gas meter.
9. Pressure of the gas at the dry gas meter.
10. Barometric pressure.
11. Readings of the dry gas meter over timed intervals of at least 3 minutes each.

Sampling. Withdraw and analyze samples of the outlet gas and the entering and exit absorbing liquor at several intervals during the run.

Typical Data. Experimental data for several runs are given in Table 2. For convenience in making the necessary calculations, the vapor

pressure of water from solutions of NaOH and Na₂CO₃ and the densities of air and carbon dioxide are given in Tables 3 and 4, respectively.

Calculations for Run No. 1.

Item 3*: *Composition of absorbing liquor, per cent Na₂O by weight*

$$\begin{aligned}
 &= \text{Item 1} \times \frac{\text{mol. wt. Na}_2\text{O}}{2 (\text{mol. wt. NaOH})} \\
 &\quad + \text{Item 2} \times \frac{\text{mol. wt. Na}_2\text{O}}{\text{mol. wt. Na}_2\text{CO}_3} \\
 &= (10.2 \times 62/80) + (0.19 \times 62/106) \\
 &= 8.0
 \end{aligned}$$

Item 5: *Carbon dioxide in exit gas, per cent by volume (wet basis)*

$$\begin{aligned}
 &= \text{Item 4} \times \frac{(p_2 - p_1)}{p_2} \\
 &= 0.47 \times \frac{(748.2 - 22.0)}{748.2} = 0.456
 \end{aligned}$$

where

p_2 = atmospheric pressure, mm. Hg = 748.2

p_1 = vapor pressure of water at 26°C. from a caustic solution containing 8.0 per cent equivalent Na₂O
= 22.0 mm. Hg (from Table 3).

Item 6: *Volume of carbon dioxide in exit gas, ft.³/hr.*

$$\begin{aligned}
 &= (\text{rate of flow of exit gas, ft.³/hr.}) \times (\text{Item 5})/100 \\
 &= 555 \times 0.456/100 = 2.53
 \end{aligned}$$

Item 11: *Volume of fresh carbon dioxide admitted, measured at temperature and pressure of outlet gas, ft.³/hr.*

$$\begin{aligned}
 &= \frac{\text{Item 7}}{\text{Item 8}} \times 60 \times \frac{T_1}{T_2} \times \frac{p_2}{\text{Item 10}} \\
 &= \frac{0.8}{2.95} \times 60 \times \frac{299.1}{299.6} \times \frac{748.2}{748.2} \\
 &= 16.25
 \end{aligned}$$

where

T_1 = absolute temperature of the outlet gas, °K.
= 273.1 + 26 = 299.1

T_2 = absolute temperature of the carbon dioxide, °K.
= 273.1 + Item 9 = 273.1 + 26.5 = 299.6

* Item numbers refer to quantities in Table 2.

TABLE 2

Experimental Data and Calculated Results for the Absorption of Carbon Dioxide by Caustic Soda

Temperature of outlet gas, °C.	= 26
Temperature of inlet liquor, °C.	= 26
Rate of flow of liquor, gal./hr.	= 288.6
Rate of flow of exit gas, ft. ³ /hr.	= 555
Barometric pressure, mm. Hg	= 748.2
Internal diameter of tower, in.	= 12
Cross-sectional area of grids, ft. ²	= 0.418
Area of wetted packing surface, ft. ²	= 51
Total absorbing area, including walls, ft. ²	= 64
Wetted perimeter of grids and walls, ft.	= 14

Item No.	Item	Run No.			
		1	2	3	4
	<i>Absorbing liquor:</i>				
1	NaOH, % by wt.	10.2	10.2	10.2	10.2
2	Na ₂ CO ₃ , % by wt.	0.19	0.19	0.19	0.19
3	Total Na ₂ O, % by wt.	8.0	8.0	8.0	8.0
	<i>CO₂ in exit gas:</i>				
4	% by volume (dry basis)	0.47	0.92	1.49	2.85
5	% by volume (wet basis)	0.456	0.894	1.45	2.77
6	Ft. ³ /hr.	2.53	4.96	8.05	15.38
	<i>Fresh CO₂ admitted:</i>				
7	Measured vol., ft. ³	0.8	1.6	2.6	5.8
8	Time of delivery, min.	2.95	4.11	4.63	6.97
9	Temp., °C.	26.5	26	26	26
10	Press., mm. Hg	748.2	747	748	748
11	Ft. ³ /hr. at temp. and press. of outlet gas	16.25	23.3	33.7	49.9
12	CO ₂ in outlet gas, % by volume	3.29	4.88	7.08	10.92
13	Density of CO ₂ at 26°C., lb./ft. ³	0.1125	0.1125	0.1125	0.1125
14	Weight fresh CO ₂ admitted, lb./hr.	1.805	2.58	3.79	5.64
	<i>CO₂ in gas stream:</i>				
15	Log-mean per cent by vol.	1.435	2.355	3.55	5.95
16	Log-mean partial press., atm.	0.01415	0.0232	0.0350	0.0584
	<i>Rate of CO₂ absorption:</i>				
17	Lb./hr.	1.805	2.58	3.79	5.64
18	Lb./(hr. x atm.)	127.6	111.3	108.3	96.6
19	Lb./(hr. x atm. x ft. ²)	1.995	1.740	1.695	1.51
20	Mols/(hr. x atm. x ft. ²)	0.0454	0.0396	0.0385	0.0343
	<i>Average rate of gas flow:</i>				
21	Ft. ³ /hr.	563	567	572	580
22	Ft./sec. through free space	0.426	0.428	0.435	0.438
	<i>Rate of liquor flow:</i>				
23	Gal. per hr. per ft. of wetted perimeter	20.6	20.6	20.6	20.6

Item 12: *Carbon dioxide in inlet gas, per cent by volume*

$$= \frac{\text{Item 6} + \text{Item 11}}{(\text{rate of flow of exit gas}) + \text{Item 11}} \times 100$$

$$= \frac{2.53 + 16.25}{555 + 16.25} \times 100 = 3.29$$

TABLE 3

Vapor Pressure of Water from Solutions of NaOH and Na₂CO₃

Temperature °C.	Per cent Na ₂ O			
	2.42	3.9	7.05	12.92
	Vapor pressure, mm. Hg			
15	14	13	11	10
18	16	15	13	12
20	18	17	16	14
22	20	19	18	16
25	24	23	21	19
27	28	27	24	21
30	33	32	30	25

TABLE 4

Density of Dry Air and Dry Carbon Dioxide at 1 Atmosphere

Temperature °C.	Density, lb./ft. ³	
	Air	CO ₂
15	0.0765	0.117
18	0.0757	0.116
20	0.0751	0.115
22	0.0746	0.114
25	0.0739	0.113
27	0.0734	0.112
30	0.0727	0.111

Item 13: *Density of CO₂ at 26°C., lb./ft.³*
 = 0.1125 (from Table 4)

Item 14: *Weight of fresh CO₂ admitted, lb./hr.*
 = Item 11 × Item 13 × pressure of exit gas, atm.
 = 16.25 × 0.1125 × 748.2/760 = 1.805

Item 15: *Log-mean per cent CO₂ in gas stream*

$$\begin{aligned}
 &= \frac{\text{Item 12} - \text{Item 5}}{2.303 \log_{10} \frac{\text{Item 12}}{\text{Item 5}}} \\
 &= \frac{3.29 - 0.456}{2.303 \log_{10} \frac{3.29}{0.456}} = 1.435
 \end{aligned}$$

Item 16: *Log-mean partial pressure of CO₂ in gas stream, atm.*

$$\begin{aligned}
 &= \frac{\text{Item 15}}{100} \times \frac{748.2}{760} \\
 &= \frac{1.435}{100} \times \frac{748.2}{760} = 0.01415
 \end{aligned}$$

Item 17: *Rate of absorption, lb. CO₂/hr.*

$$= \text{Item 14} = 1.805$$

(As the outlet gas was circulated, all of the fresh CO₂ admitted was absorbed).

Item 18: *Rate of absorption, lb. CO₂/(hr. × atm.)*

$$= \frac{\text{Item 17}}{\text{Item 16}} = \frac{1.805}{0.01415} = 127.6$$

Item 19: *Rate of absorption, lb. CO₂/(hr. × atm. × ft.²)*

$$= \frac{\text{Item 18}}{\text{total absorbing area, ft.².}} = \frac{127.6}{64} = 1.995$$

Item 20: *Rate of absorption, mols CO₂/(hr. × atm. × ft.²)*

$$= \frac{\text{Item 19}}{\text{mol. wt. CO}_2} = \frac{1.995}{44} = 0.0454$$

Item 21: *Average rate of gas flow, ft.³/hr.*

$$\begin{aligned}
 &= \text{exit gas, ft.³/hr.} + \frac{1}{2} \text{Item 11} \\
 &= 555 + 16.25/2 = 563
 \end{aligned}$$

Item 22: *Velocity of gas through free space, ft./sec.*

$$\begin{aligned}
 &= \text{Item 21} \times 1/3600 \times 1/(A_t - A_p) \\
 &= 563 \times 1/3600 \times 1/(0.786 - 0.418) \\
 &= 0.426
 \end{aligned}$$

where

$$\begin{aligned}
 A_t &= \text{cross-sectional area of tower, ft.}^2 \\
 &= \frac{\pi \times 12^2}{144 \times 4} \\
 &= 0.786 \\
 A_g &= \text{cross-sectional area of grids, ft.}^2 \\
 &= 0.418
 \end{aligned}$$

Item 23: *Rate of flow of liquor, gal. per hr. per ft. of wetted perimeter*
 $= 288.6/14 = 20.6$

Discussion of Results. The results of this series of runs indicate that there is a definite decrease in the unit rate of absorption with increase in the concentration of the gas. This may be explained as follows:

1. The absorption of carbon dioxide in a solution of sodium hydroxide is not a simple, single reaction but occurs in several successive stages:

- a. $\text{CO}_2 (\text{gas}) \rightleftharpoons \text{CO}_2 (\text{dissolved})$
- b. $\text{CO}_2 (\text{dissolved}) + \text{H}_2\text{O} \rightleftharpoons \text{H}_2\text{CO}_3$
- c. $\text{H}_2\text{CO}_3 + \text{NaOH} = \text{NaHCO}_3 + \text{H}_2\text{O}$
- d. $\text{NaHCO}_3 + \text{NaOH} = \text{Na}_2\text{CO}_3 + \text{H}_2\text{O}$
- e. $\text{Na}_2\text{CO}_3 + \text{H}_2\text{CO}_3 = 2\text{NaHCO}_3$

2. Of these reactions, the absorption of CO_2 , as such, in water, and the interaction between CO_2 and H_2O to form H_2CO_3 are reversible. Unless the carbonic acid is rapidly and completely removed from the interfacial zone by interaction with NaOH or Na_2CO_3 , a considerable back pressure of carbon dioxide will develop and the rate of absorption will decrease.

3. At very low concentrations of carbon dioxide, the sodium hydroxide diffuses, or is carried by convection, to the interface fast enough to react with and remove the carbonic acid as fast as it is formed. Thus, under such conditions, the overall rate of absorption depends on the rate at which the carbon dioxide is brought to the interface and the rate at which carbon dioxide in solution reacts with the water to form carbonic acid. At very low concentrations, this latter factor becomes negligible, because even the carbon dioxide present, as such, in solution will be carried away from the surface as fast as it dissolves.

4. As the concentration of carbon dioxide is increased, the rate of access of alkali to the surface (and the rate of interaction of dissolved CO_2 with the water) becomes of increasing importance. As a result, the liquid film becomes more and more important and the overall rate of absorption shows a slight but definite decrease.

In most cases, the change in the overall absorption rate with change in logarithmic partial pressure of carbon dioxide in the gas is at least approximately linear. By plotting the absorption rates (Item 20) as ordinates and the concentrations (Item 15) as abscissas and extrapolating, a value can be found for the overall rate at a limiting concentration of zero per cent carbon dioxide in the gas, i.e., the value for the overall rate when the conditions are such that the overall resistance is equal to the resistance of the gas film. A limiting value of 0.0435 mols per hour per square foot at zero partial pressure was obtained from the data in this series of experiments. The corresponding value for the average rate of gas flow was found, by plotting Item 21 against Item 15, to be 555 cubic feet per hour, which corresponds to a velocity of 0.419 feet per second through the free space.

SECTION 7

FILTRATION

INTRODUCTION

The mechanical separation of a solid material from a liquid in which it is suspended can be accomplished by several different methods, but the most important of these in the chemical process industries is filtration. In this method, the separation is effected by a porous medium which permits the passage of the liquid but retains the solid particles.

Although filtration is generally regarded as an operation for the recovery of either the solid, the liquid, or both, this does not constitute the only objective of commercial filtration. In the chemical process industries, the majority of filters are utilized not only to effect the desired separation but also to wash soluble materials from the deposited cake. In some processes, on the other hand, the principal objective of filtration is to clarify a liquid that contains a small quantity of suspended matter. For efficient operation at minimum cost, therefore, it is necessary to select a filter that is best suited for its intended purpose.¹⁰:

Types of Filters. Filters may be classified broadly into three general types: (1) gravity filters, (2) pressure filters, and (3) vacuum filters. Filters of the first type, of which open sand filters are representative, find but little use in the chemical process industries, but the other two types are used extensively.

Pressure Filters. These are built in a number of different styles, but the most important are the *chamber*, and the *plate and frame* filter presses. A press of either type consists of a framework made of two end supports connected by two horizontal, parallel bars on which a varying number of filter cells are assembled. These cells are formed either by cloth-covered, recessed filter plates—chamber filter press—or by alternate frames and cloth-covered plates—plate and frame press. The press is closed by means of a screw or a hydraulic ram, which holds the cells tightly together. The slurry is forced into the cells, which are so designed that the liquid cannot leave the press except by passing through the cloth membranes.

Plate and frame presses have a number of advantages over the chamber press. In the latter type, the cloth is subjected to strain, and the wear is, therefore, severe; furthermore only relatively thin cakes can be built up, and the construction does not permit satisfactory washing of the cake.¹

In addition to the plate and frame types, pressure filters are also constructed in the form of *leaf filters*, in which a number of membrane-covered leaves are held within a shell into which the slurry is charged under pressure; the liquid passes through the membranes while the solids deposit on the outside surface.

Vacuum Filters. The simplest type of vacuum filter consists of a false-bottomed tank, the bottom outlet of which is connected to a source of vacuum through a liquor receiver. Although this type of filter is relatively inexpensive and easy to operate, its capacity is very low.

Where large quantities of slurry are to be filtered, the *leaf type of vacuum filter*, or the *rotary continuous vacuum filter* is generally used. In the former, a large number of filter leaves, attached to a common header, are suspended in a tank containing the slurry; vacuum is then applied through the header. When the desired thickness of cake has been deposited, the leaves are lifted mechanically from the tank, and the cakes are either discharged directly by blowing compressed air through the header, or they are first washed by placing them in a washing tank and drawing water through them.

Rotary continuous vacuum filters are especially suited for the filtering of large quantities of materials that form cakes rapidly. In these filters, a large rotating drum serves as the support for the filter medium; in one type—external-drum vacuum filter—the filter medium covers the outside surface of the drum, and in the second type—internal-drum vacuum filter—the medium covers the inside surface. In both types, the drum is divided into separate compartments, which are automatically connected, through valves, alternately with suction and with compressed air during the rotation of the drum.

Materials of Construction of Chamber, and Plate and Frame Presses. Pressure filters of the chamber, and the plate and frame types are constructed from a number of different materials, including cast iron, lead, bronze, aluminum, wood, rubber, stainless steel, and special alloys; of these, however, cast iron is by far the most commonly used because of its relative cheapness, its strength, and its durability.

The Filter Membrane. Filter membranes are made, principally, from a number of different woven materials, including cotton, wool, animal and human hair, metal, asbestos, and glass. Of these, cotton finds the most widespread use because it is mechanically strong and relatively inexpensive, but it cannot be used for filtration at high temperatures or with solutions of very high or very low pH because such

solutions are corrosive to cotton. The next most important fiber for filter cloth is wool, which is more acid-resistant than cotton and is capable of withstanding considerably higher temperatures without undergoing structural breakdown; however, wool cannot be used with alkaline solutions.

The filtration characteristics of a woven filter membrane are controlled by: (1) the weight of the cloth; (2) the thread count, that is, the number of strands per inch, and (3) the style of weave. The three principal weave styles are twill, chain, and plain. The weave determines the porosity and the flow capacity of the cloth; for cloths made of an equal number of strands of the same diameter threads, minimum porosity and flow capacity are obtained with the plain weave, while the highest porosity and capacity are provided by the chain weave.⁶

Filter Aids. Slurries of finely divided and gelatinous materials are often difficult to filter because they plug the pores of the filter medium. This increases the resistance to the flow of liquor and results in a marked decrease in the filtration rate. This difficulty may be overcome by adding to the slurry a solid material capable of building up a rigid skeleton which is sufficiently porous to permit the free passage of the liquid and yet retain the finely-divided solid particles. Such materials, which are known as filter aids, must be chemically inert to the slurry and, in order to insure their remaining in suspension when mixed with the slurry, they should have a low specific gravity.

A number of different substances are utilized for this purpose; these include brick dust, precipitated calcium carbonate, paper pulp, and numerous other substances, but the most widely used material is kieselguhr, or diatomaceous earth.³ To be effective, the filter aid must be correctly proportioned, for if too much is used the increased thickness of the cake tends to counterbalance the advantage of increased porosity.

Laboratory Filter Presses. Representative small-scale filters of many of the commonly employed commercial units can be purchased from manufacturers. Such filters are valuable in acquainting the student with the mechanical features of their operation, and also find an important place in pilot plant work.

Filtration experiments, however, do not require an elaborate set-up. Generally, a laboratory press with a single plate and frame, or a single-leaf filter is sufficient to obtain all of the necessary test data. A satisfactory arrangement for filtration experiments at constant pressure and constant rate is shown in Figure 1, page 440; and a set-up for vacuum filtration tests is shown in Figure 1, page 456.

LABORATORY FILTRATION EXPERIMENTS

Filtration experiments are designed, principally, to obtain data from which the so-called filtration constants for the material being filtered may be calculated. These constants, in turn, may then be used to calculate such design factors as: (1) the filtering area required for a given capacity when operating at a specified filtering pressure; (2) the time required for washing; (3) the optimum frame thickness for maximum overall capacity, taking into consideration the time required for filtering, washing, and dressing the press; and (4) the optimum frame thickness for minimum filtration cost.

Filtration constants may be determined by: (1) filtering the slurry at a constant pressure; (2) filtering the slurry at a constant rate; (3) passing water through a bed or cake of the solids; (4) filtering the slurry at a constant pressure-time gradient, in which the pressure is increased by equal small increments at uniform intervals of time; and (5) filtering the slurry at a constant pressure-volume gradient, in which the pressure is increased by equal increments at equal increments of filtrate volume.⁷ Most filtration tests are carried out by the first method, but considerable attention is now being given to methods 2 and 3. The last two methods are more difficult to control and have, as yet, found little use.

Analysis of Constant-Pressure Filtration Data. The resistance offered to the passage of the filtrate consists of: (1) the resistance of the filter press leads and channels, (2) the resistance of the cake, and (3) the resistance of the filter medium. The sum of these resistances may be related to the differential rate of filtrate flow in the following manner:

$$\frac{dV}{d\theta} = \frac{A}{\mu} \frac{P}{R} \quad (7.1)$$

where

V = volume of filtrate collected up to the time θ .

θ = elapsed time.

A = total filtering area.

μ = viscosity of the filtrate.

P = pressure drop through the press.

R = total resistance to the flow of the filtrate.

If the press is designed properly, the resistance of the leads and the channels will be negligible and, therefore, the resistance to the flow of filtrate may be considered to be made up of the resistances offered by the cake and the filter medium. The total resistance offered by the

cake increases as the cake increases in thickness, and depends upon the character of the slurry, the filtration rate, and the pressure. Furthermore, the resistance of the filter medium* is not constant but also depends upon the rate, the pressure, and the character of the slurry.

The Ruth Equation. To simplify the handling of these resistances, Ruth and his co-workers³ developed an equation in which the resistance of the filter medium is expressed in terms of an equivalent thickness of cake; this equation is

$$(V + C)^2 = K(\theta + \theta_0) \quad (7.2)$$

where

V = total volume of filtrate delivered, ft.³

C = volume of filtrate required to produce a cake thickness equal in resistance to that of the filter medium, ft.³

θ = time of filtration, hr.

θ_0 = time required to form a cake having a resistance equal to the resistance of the filter medium, hr.

K = a constant, which is a function of the filtration pressure and is related to it in the following manner:

$$K = \frac{2A^2P(1 - ms)}{\mu\rho s\alpha} \quad (7.3)$$

where

A = filter area, ft.²

P = filtration pressure, i.e., pressure drop through press, lb./ft.²

m = lb. of wet cake per lb. of filterable solids (dry cake).

s = weight fraction of filterable solids in the slurry.

μ = viscosity of filtrate, lb./(ft. \times hr.).

ρ = density of filtrate, lb./ft.³

α = average specific resistance of the cake

= resistance of one lb. of dry solid deposited upon one ft.² of filtering area, hr.²/lb.

Determination of Constants K , C , and θ_0 . When differentiated with respect to V , Equation (7.2) becomes

$$\frac{d\theta}{dV} = \frac{2V}{K} + \frac{2C}{K} \quad (7.4)$$

*The true medium is the filter cloth partially plugged with particles of solids from the slurry.

Since C is a constant, if values of $d\theta/dV$ are plotted against values of V , the resulting curve will be a straight line with a slope of $2/K$, an intercept on the $d\theta/dV$ axis of $2C/K$, and a negative intercept on the V axis of C . The slope $2/K$ is a measure of the filter cake resistance, and the intercept C is a measure of the filter medium resistance under the conditions of the experiment.

The values of $d\theta/dV$ for corresponding values of V , needed to plot Equation (7.4), can be obtained by plotting the experimentally determined values of V versus θ and either graphically differentiating the curve or determining the slopes at various points by means of a tangentiometer.

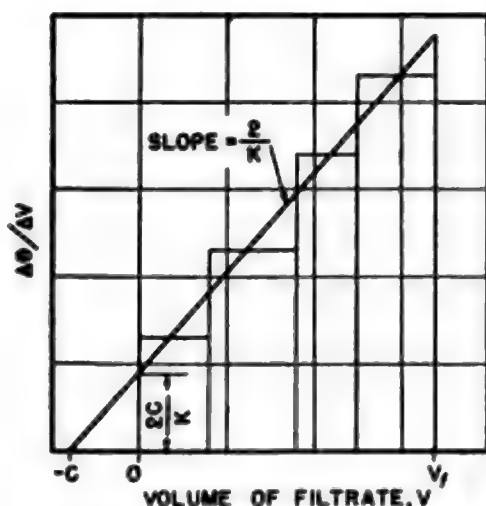


Figure 1. Plot of Constant Pressure Filtration Data According to the

$$\text{Equation: } \frac{d\theta}{dV} = \frac{2V}{K} + \frac{2C}{K}.$$

For most purposes, however, it is sufficiently accurate to plot values of $\Delta\theta/\Delta V$ against V and draw the best straight line through the midpoints, as shown in Figure 1. The term ΔV represents the quantity of filtrate collected over a finite interval of time $\Delta\theta$.

Having determined the values of K and C , the values of θ_0 can be calculated by means of Equation (7.2), which may be rewritten as

$$\theta_0 = \frac{(V + C)^2}{K} - \theta \quad (7.5)$$

In using this equation, several different values of V are substituted into the equation, and a mean value of θ_0 is determined by averaging the results. The mean value obtained in this manner will usually be found to differ by a few seconds from the value calculated by setting V and θ equal to zero in Equation (7.5), which then becomes

$$C^2 = K\theta_0 \quad (7.6)$$

This discrepancy is usually due to the time lag in most filtration experiments between the start of the filtration and the time at which the desired constant pressure is reached. This time lag, however, may be reduced to zero by providing pet cocks at the top of each frame when using a plate and frame filter press setup.⁵ The following procedure is then employed: the press and filtrate lines are completely filled with water previous to the start of a test; the pressure on the slurry supply

is adjusted to the desired value; and, with the filtrate outlet closed and the pet cocks on the frames open, the slurry is passed through the system to completely displace the water in the frames. The filtration experiment is then started immediately by closing the pet cocks and opening the filtrate outlet and the slurry supply valves instantaneously with the start of a stop watch.

Determination of Cake Resistance α and Relative Compressibility n . The average specific resistance of the filter cake can be calculated by means of Equation (7.3), which may be rewritten as

$$\alpha = \frac{2A^2 P(1 - ms)}{K\mu\rho s} \quad (7.7)$$

If runs at more than one pressure are made, values of P may be plotted on log-log paper against corresponding calculated values of α ; this gives a straight line, the general equation of which is

$$\alpha = \alpha_0 + \gamma P^n \quad (7.8)$$

where

α_0 , γ and n are constants.

The slope of the curve, n , obtained from such a plot, is a measure of the compressibility of the cake; for extremely compressible substances, the value of n is about 0.8; for substances of medium compressibility it is about 0.5; and as the degree of compressibility decreases still further it approaches zero.³

The Lewis Equation. Walker, Lewis, McAdams, and Gilliland³ give the following filtration equation

$$\frac{dV}{d\theta} = \frac{PA^2}{\mu(r''vP^*V + \rho'AP^m)} \quad (7.9)$$

Although this equation appears to be materially different from Equation (7.2), Larian⁴ has shown that the two equations are essentially similar.

When integrated for constant pressure, Equation (7.9) becomes

$$\frac{P\theta}{(V/A)} = \frac{r''v\mu}{2} P^* \left(\frac{V}{A} \right) + \rho'\mu P^m \quad (7.10)$$

where

P = pressure drop across cake, filter medium, and press channels.

θ = time.

V = weight of filtrate up to time θ .

A = filtering area.

r'' = cake resistivity.

v = volume of cake per unit of filtrate.

μ = viscosity of filtrate.

s = cake resistivity exponent.

ρ' = cloth resistivity.

m = cloth resistivity exponent.

When carrying out an experiment, the quantity of filtrate collected is not ordinarily equal to the total weight of filtrate V because of the holdup in the filtrate line. If the filtrate holdup is V_c and the filtrate collected up to time θ is V' , the total weight of filtrate up to time θ is $V' + V_c$, and Equation (7.10) becomes

$$\frac{P\theta}{(V' + V_c)/A} = \frac{r''v\mu}{2} P^s \frac{(V' + V_c)}{A} + \rho'\mu P^m \quad (7.11)$$

Furthermore, because there is usually a time lag before the constant pressure is reached, an additional correction must be applied. If this time lag is equal to θ_r and the quantity of filtrate collected during this period is V_r , the integration of Equation (7.9) for the period after the constant pressure is reached is as follows:

$$\int_{V_c+V_r}^{V'+V_c} \mu(r''vP^sV + \rho'AP^m) dV = \int_{\theta_r}^{\theta} PA^2 d\theta \quad (7.12)$$

or

$$(\mu r''vP^s) \int_{V_c+V_r}^{V'+V_c} V dV + (\mu \rho'AP^m) \int_{V_c+V_r}^{V'+V_c} dV = PA^2 \int_{\theta_r}^{\theta} d\theta \quad (7.13)$$

which integrates to

$$\begin{aligned} \mu r''vP^s \left[\frac{(V' + V_c)^2 - (V_c + V_r)^2}{2} \right] \\ + \mu \rho'AP^m [(V' + V_c) - (V_c + V_r)] = PA^2(\theta - \theta_r) \end{aligned} \quad (7.14)$$

and simplifies to

$$\begin{aligned} \mu r''vP^s \frac{(V' + V_r + 2V_c)(V' - V_r)}{2} \\ + \mu \rho'AP^m(V' - V_r) = PA^2(\theta - \theta_r) \end{aligned} \quad (7.15)$$

or

$$\frac{PA(\theta - \theta_r)}{V' - V_r} = \frac{\mu r''vP^s(V' + V_r + 2V_c)}{2A} + \mu \rho'P^m \quad (7.16)$$

The Simplified Filtration Equation. If the resistance of the filtering medium is assumed to be small and is neglected, Equation (7.2) may be written as

$$V^2 = K\theta \quad (7.17)$$

which is the equation of a straight line passing through the origin and having a slope equal to K . But, when experimental values of V^2 are plotted against θ , the curve generally has an S-shape, as shown in Figure 2, and no definite value of K can be determined. However, Larian⁴ has shown that if a straight line is drawn through the origin and the terminal point of the experimental S-shaped curve and the slope of this straight line is taken as the value of K for use in Equation (7.17), the results are sufficiently accurate for most industrial filtration experiments.

The degree of compressibility n of the cake can then be determined by making several runs at different pressures, calculating the values of K for each test by means of Equation (7.17), and then plotting the values of P against the values of K on log-log paper; this gives a straight line having a slope of $1 - n$.

Rate of Washing. The rate of washing can be calculated from the rate of filtration at the end of the filtration cycle which, in turn, may be determined by differentiating Equation (7.2) with respect to time

$$\frac{dV}{d\theta} = \frac{K}{2(V_f + C)} \quad (7.18)$$

where V_f is the total volume of filtrate during the filtration cycle.

If it may be assumed that: (1) the rate of washing is constant throughout the washing period, (2) the wash water follows the same path as the filtrate, (3) the wash water passes through the same thickness of cake as the final filtrate, and (4) the viscosity of the wash water is the same as that of the filtrate, Equation (7.18) also gives the rate of washing.

The first three assumptions hold rather closely for leaf filters, but in a plate and frame filter press the wash water passes from one plate completely through the cake and discharges through an adjacent plate;

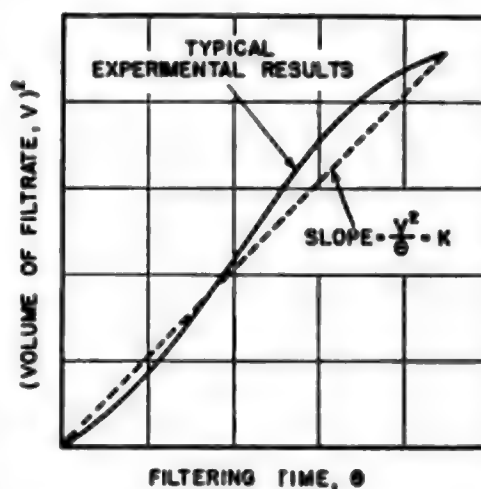


Figure 2. Plot of Constant Pressure Filtration Data According to the Equation $V^2 = K\theta$

therefore, the wash water passes through twice the thickness of cake traversed by the final quantity of filtrate, and the area available for washing is only one-half of that used during filtering. Under these conditions, the rate of washing is approximately one-fourth the final rate of filtration.

If the viscosity of the wash water differs from that of the filtrate, this fact must be taken into consideration. It is evident from Equation (7.3) that K is inversely proportional to viscosity; therefore, the washing rate is also inversely proportional to the viscosity.

Since in most cases C is very small compared to V_f , for practical purposes Equation (7.18) may be written as

$$\frac{dV}{d\theta} = \frac{K}{2V_f} \quad (7.19)$$

or, if the viscosity is taken into consideration, as

$$\frac{dV}{d\theta} = \frac{K}{2V_f} \times \frac{\mu_s}{\mu_w} \quad (\text{for leaf filters}) \quad (7.20)$$

or

$$\frac{dV}{d\theta} = \frac{K}{8V_f} \times \frac{\mu_s}{\mu_w} \quad (\text{for plate and frame filters}) \quad (7.21)$$

where μ_s and μ_w are the viscosities of the filtrate and the wash water, respectively.

Analysis of Constant Rate Filtration Data. The analysis of constant-rate filtration data is much more complicated than the corresponding treatment of constant pressure data. However, constant rate data can be analyzed by either the *Ruth* equation—(7.2)—or by the *Lewis* equation—(7.9).

Since for constant rate $dV/d\theta$ is constant and, therefore, equal to V/θ , Equation (7.9) becomes

$$\frac{V}{\theta} = \frac{PA^2}{\mu(r''vP^sV + \rho'AP^m)} \quad (7.22)$$

or

$$\frac{P\theta}{(V/A)} = r''v\mu P^s \left(\frac{V}{A} \right) + \rho'\mu P^m \quad (7.23)$$

The procedure for handling constant rate data has been outlined by Bonilla,³ and it is illustrated in his report on constant rate filtration—pages 448 to 454.

NOMENCLATURE

- A = total filtering area.
 C = quantity of filtrate required to produce a cake thickness equal in resistance to that of the filter medium.
 K = a constant.
 m = weight ratio of wet cake to dry cake in Ruth equation, and cloth resistivity exponent in Lewis equation.
 n = measure of relative compressibility of cake.
 P = pressure drop through press.
 r'' = cake resistivity.
 R = total resistance to the flow of filtrate.
 s = weight fraction of filterable solids in slurry in Ruth equation, and cake resistivity exponent in Lewis equation.
 v = volume of cake per unit of filtrate.
 V = quantity of filtrate.
 V' = quantity of filtrate collected up to time θ
 = total quantity of filtrate V minus the filtrate-line holdup V_c .
 V_c = quantity of filtrate held up in filtrate line.
 V_f = total quantity of filtrate during the filtration cycle.
 V_r = quantity of filtrate collected before desired constant pressure is reached.
 ΔV = quantity of filtrate collected over finite time interval $\Delta\theta$.
 α = average specific resistance of cake.
 α_0 = a constant.
 γ = a constant.
 θ = elapsed time.
 θ_0 = time required to form a cake having a resistance equal to the resistance of the filter medium.
 θ_r = elapsed time before desired constant pressure is reached.
 μ or μ_f = viscosity of filtrate.
 μ_w = viscosity of wash water.
 ρ = density of filtrate.
 ρ' = cloth resistivity.

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A TEST SETUP FOR FILTRATION AT CONSTANT PRESSURE AND AT CONSTANT RATE

Designed and Constructed by

The Department of Chemical and Gas Engineering
Johns Hopkins University*

Description. This setup, shown in Figure 1, includes: (1) a 12-inch washing plate and frame filter press, (2) a 300-gallon wood-stave slurry tank provided with an agitator, (3) a 10-gallon per minute rotary pump with constant speed drive, (4) a by-pass return line from the pump to the tank to regulate the filtration pressure or the flow rate, and (5) an orifice meter in the filtrate line to aid in adjusting the rate of flow during constant rate runs.

Materials and Cost. The approximate cost of the materials and equipment required for this setup is \$250. A list of these items is given in Table 1.

Auxiliary Equipment. In addition to the setup shown in Figure 1, the following auxiliary equipment is needed to carry out an experiment:

1. Thermometer: 0 to 100°C. in 1°C.
2. Stop watch: 60 seconds in $\frac{1}{2}$ second for 2 hour.
3. Platform scale: 500-pound capacity, 50-pound beam graduated in 2 pound.
4. Scale or balance: graduated in grams.
5. Can or large bucket to receive the filtrate.
6. Large spatula, tin pans, and drying oven.

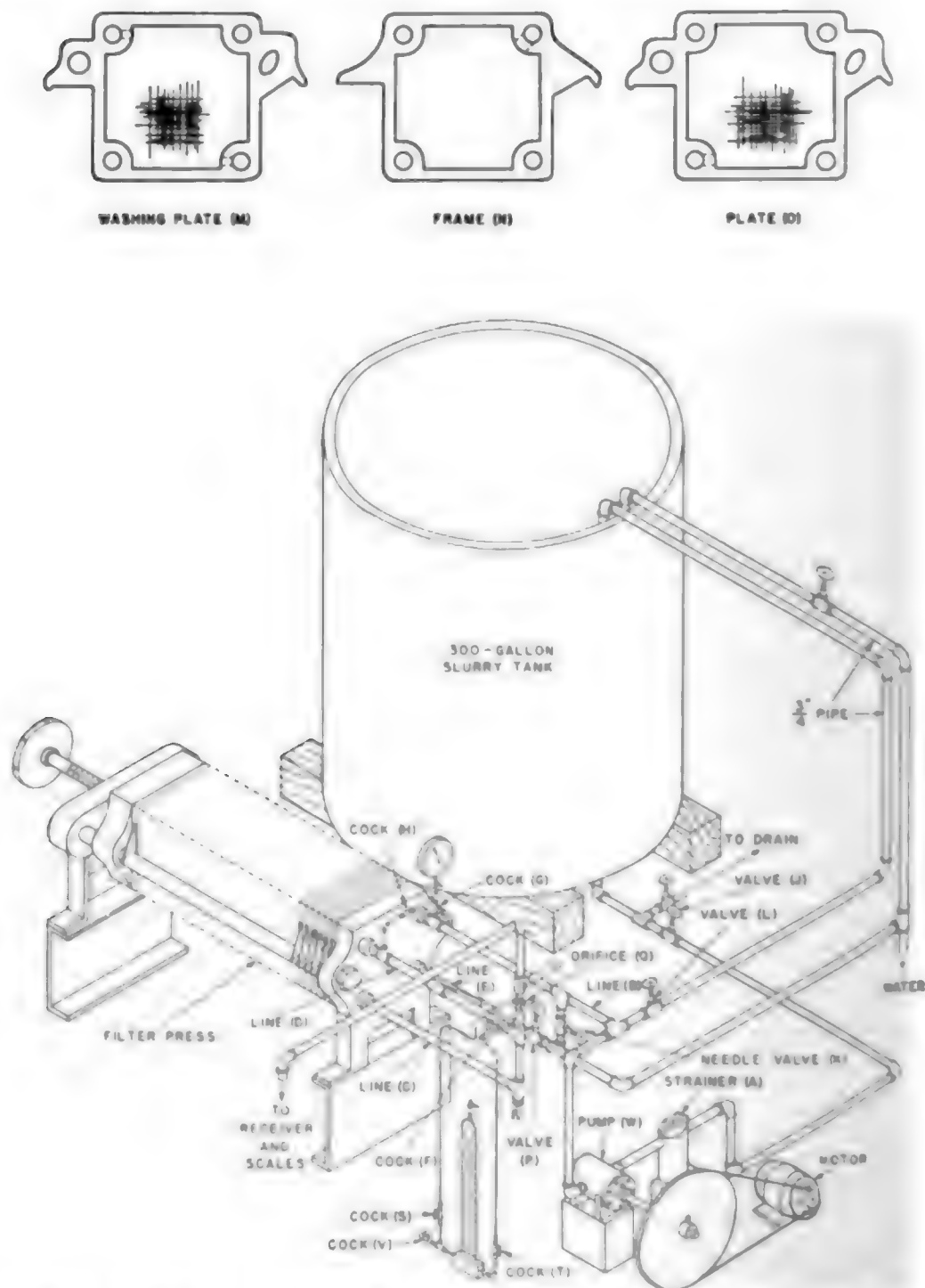
Notes on Construction. 1. Valve *L*, in the by-pass line, should be so located that it can be conveniently controlled by the operator while watching either the pressure gage or the orifice manometer.

2. Orifice *Q*, in the filtrate line, need not be standard as it is only used as an aid in keeping the flow rate constant during constant-rate runs. It may be a gasket-type union in which any one of a number of plates with different sized orifices may be inserted so that manometer readings of 4 to 12 inches of water under air may be obtained over a considerable range of flow rates.

3. Pump *W* and suction-line strainer *A* should be readily accessible for dismounting and cleaning.

4. A pet cock is provided at the low point in the filtrate line to make it possible to drain this line at the end of the filtration experiment so that

* Reported by C. F. Bonilla



the retained liquor can be weighed and the weight can be applied as a correction to the weight of the bulk of the filtrate which is collected. However, if preferred, this correction may be calculated.

TABLE 1
Bill of Materials for Filtration Setup
Johns Hopkins University

Quantity	Description	Size	Material
45 ft.	Std. pipe	$\frac{1}{2}$ in.	Galv. wrought steel
25	Std. nipple	$\frac{1}{2}$ in., assorted lengths	" " "
11	" "	$\frac{1}{2}$ in., assorted lengths	" " "
9	Std. tee	$\frac{1}{2}$ in.	Galv. malleable iron
2	" "	$\frac{1}{2} \times \frac{1}{2} \times \frac{1}{2}$ in.	" " "
2	" "	$\frac{1}{2}$ in.	" " "
20	Std. 90° elbow	$\frac{1}{2}$ in.	" " "
2	" " "	$\frac{1}{2}$ in.	" " "
1	Std. gasket-type union	$\frac{1}{2}$ in.	" " "
7	Ground-joint union	$\frac{1}{2}$ in.	" " "
1	" " "	$\frac{1}{2}$ in.	" " "
1	Bushing	$\frac{1}{2} \times \frac{1}{2}$ in.	Cast iron
2	"	$\frac{1}{2} \times \frac{1}{2}$ in.	Malleable iron
3	Globe valve, disc seat	$\frac{1}{2}$ in.	Brass
3	Needle-point valve	$\frac{1}{2}$ in.	"
1	Three-way cock	$\frac{1}{2}$ in.	"
1	Two-way cock	$\frac{1}{2}$ in.	"
1	Filter press, closed discharge, washing-type, with four frames and dummy end plate	12 in.	Cast iron
1	Round wood tank	4-ft. diam., 300 gal.	2-inch fir
1	Mixer (propeller type)	$\frac{1}{2}$ H.P.	Stainless steel shaft
1	Quick-opening strainer with removable basket	3 x 4 in.; 20 mesh	Cast iron, with brass basket
1	Pressure gage	0 to 100 lb. press., 5 in. or larger face	Steel and brass
1	Manometer, inverted type	16 in. long	Glass
6	Gage cock	$\frac{1}{2}$ in.	Brass
1	Pet cock	$\frac{1}{2}$ in. male I.P.S.	"
15 ft.	Tubing	$\frac{1}{8}$ in. O.D.	Copper
4	Compression-type fitting	$\frac{1}{8}$ in. copper to $\frac{1}{2}$ in. male I.P.S.	Brass
10 yds.	Filter cloth	16.5 oz./yd.	Cotton; twill weave
1	Rotary pump (vane type)	10 gal./min.	Steel or bronze
1	Electric motor	$\frac{1}{2}$ or $\frac{1}{4}$ H.P.	—
1	Motor starter	$\frac{1}{2}$ or $\frac{1}{4}$ H.P.	—
1	V-belt	$\frac{1}{2}$ in.	Rubberized cotton

5. It is desirable to have a drainage trough under the filter press for convenience when hosing down the press and floor and for emptying filtrate receivers.

Comments on Design. 1. A variable-speed drive or motor may be used instead of a by-pass line to control the filtration pressure and rate.

2. For most tests, a maximum of four frames is sufficient; normally only one or two frames are employed.

3. The thickness of the filter frames is a matter of individual preference; one-inch frames give longer and more accurate runs and permit the use of higher pressures, while $\frac{1}{2}$ - or $\frac{3}{4}$ -inch frames can be filled and washed more rapidly, thus reducing the time for an experiment. Furthermore, the smaller cakes can be dried much more rapidly, thus expediting moisture-content determinations.

Operation. Slurry, mixed or precipitated in the tank, passes through strainer *A* and pump *W*, and enters the press through line *B*; some of it returns to the tank through valve *L*, in the by-pass line. The filtrate enters the frames *N*, and leaves the end plates and even numbered plates *O* through line *C* and the odd numbered or washing plates *M* through lines *D* and *E*; the streams finally combine and flow to the receiver. During this operation, valve *P* and cock *H* are closed and cock *F* is open.

During washing, cocks *G* and *F* and valves *L* and *J* are closed and cock *H* is open. Water from the main enters the press through valve *P* and line *D*, leaves through line *C*, and passes into the receiver where it is weighed.

To measure the cloth resistance, cocks *F* and *H* and valves *J*, *L*, and *P* are closed. The water is passed through needle valve *K*, enters the press through line *B*, leaves through line *C*, and passes into the receiver where it is weighed.

TYPICAL CONSTANT-PRESSURE FILTRATION EXPERIMENT

Objects. To study the filtration characteristics of a slurry of precipitated chalk at constant pressure, and to determine the resistance of the filter cloth.

Procedure. 1. Prepare 250 gallons of an aqueous slurry of about 1 to 2 per cent of precipitated chalk in the make-up tank.

2. Carry out three or more filtration runs with this slurry at different pressures using one frame in the press, and then carry out another run with a washing plate and two frames and measure the washing rate at several water pressures with full cakes. After the last run, open the

press, remove the cakes, and scrape the cloths clean but do not wash them. Replace the cloths, and determine the cloth resistance by measuring the rate of flow of water at several different pressures.

3. In making a run, place the wetted cloth in the press, keeping loose threads away from the bearing surfaces. When the press is ready, start the pump with by-pass valve *L* open, and then quickly close this valve and pump the slurry through the press, receiving the filtrate in a tared can. When the desired pressure has been reached, slowly open valve *L* so as to keep the filtration pressure constant, and almost close cock *G* so that the needle of the pressure gage does not vibrate. Continue the filtration until the frame is filled and the rate drops rapidly.

4. During each run, record: (a) the time and filtrate weight at which the desired pressure is reached*; (b) the time and weight of filtrate at even increments of filtrate, such as every 10 pounds, as the scale arm rises at the pre-set weight; and (c) the average filtrate temperature.

5. At the end of the run, open the press and examine the cake. Transfer the cake to a tared, shallow tin pan, weigh it, dry it in an oven at 110°C., and then reweigh it.

6. Measure the difference in elevation between the center of the gage and the end of the filtrate-discharge pipe to obtain the correction that is to be applied to the reading of the pressure gage.

7. Analyze the data obtained in these experiments by means of the constant-rate equation, and calculate the values of the cake and cloth resistances.

Experimental Results. The experimental data for two filtration runs are given in Table 2, and the experimental data for four separate water runs for the determination of the filter cloth resistance are given in Table 3.

Analysis of the Data. The equation† for filtration at constant pressure is³

$$\frac{P(\theta - \theta_r)A}{V - V_r} = \frac{r''v}{a} P \frac{\mu a(V + V_r + 2V_c)}{2A} + \mu \rho' P^m \quad (1)$$

where

P = pressure differential across cloth and cake, lb./in.²

θ = total time of filtering, min.

* For accuracy of results, the desired pressure should be reached as soon as possible after the experiment is started.

† The derivation of this equation is shown on page 434. It should be noted, however, that the *V* in Equation (1) corresponds to the *V'* in Equation (7.16).

TABLE 2

*Experimental Results for Filtration of CaCO₃ at Constant Pressure*Filter Cloth: Twill weave, 20/3 ply warp, 8/3 ply filling, 16.5 oz./yd.²

Slurry: Precipitated chalk, 1 per cent (water basis).

Area of one frame-side, in.²: $(9.625)^2 - 3.14(1.5)^2 = 85.6$.Filtrate holdup correction, $V_r = 2$ lb. for 1 frame and 2.5 lb. for 2 frames.Pressure correction*, lb./in.² = -0.3.

Run Number.....	1		2	
No. of frames used.....	2		1	
Gage reading, lb./in. ²	10		25	
Temperature, °C.....	20.5		20.5	
Viscosity of filtrate, centipoises.....	1.0		1.0	
Reading number	Filtering time θ , min:sec.	Weight of filtrate V , lb.	Filtering time θ , min:sec.	Weight of filtrate V , lb.
1	0:13 (θ_r)	7.75 (V_r)	0:43 (θ_r)	25.5 (V_r)
2	0:40	27.75	1:22	45.5
3	1:10	47.75	2:14	65.5
4	2:10	77.75	3:18	85.5
5	2:57	97.75	4:35	105.5
6	3:51	117.75	5:06	125.5
7	4:54	137.75	7:41	145.5
8	6:03	157.75	9:36	165.5
9	7:22	177.75	11:46	185.5
10	—	—	14:15	205.5

* The filtrate discharge is 8 inches below the center of the pressure gage.

TABLE 3

Experimental Data and Calculated Results of Water Test for Filter Cloth Resistance

Item	Run No.		Average of Runs 1 and 2	Run No.		Average of Runs 3 and 4
	1	2		3	4	
Temperature, °C.....	22	22	22	22	22	22
Viscosity of water, centipoises.....	0.96	0.96	0.96	0.96	0.96	0.96
Pressure, lb./in. ²	5	5	5	10	10	10
Time for 10 lbs. of water, sec.....	38.0	39.0	38.5	21.8	21.6	21.7
Cloth resistance, P^m			53.8			62.4

 θ_r = time of filtering before the desired constant pressure is reached, min. A = cloth area, in.² V = total weight of filtrate collected, lb. V_r = weight of filtrate collected during θ_r , lb.

r'' = cake resistivity.

v = volume of cake, in.³/lb. filtrate.

a = concentration of slurry, lb. solid/100 lb. filtrate.

s = cake resistivity exponent.

μ = viscosity of filtrate, centipoises.

V_c = weight of filtrate trapped in the filtrate line, lb.

ρ' = cloth resistivity.

m = cloth resistivity exponent.

1. From the data for each filtration run after the desired pressure was reached, values of $\frac{P(\theta - \theta_r)A}{V - V_r}$ and $\frac{(V + V_r + 2V_c)}{A}$ were calculated

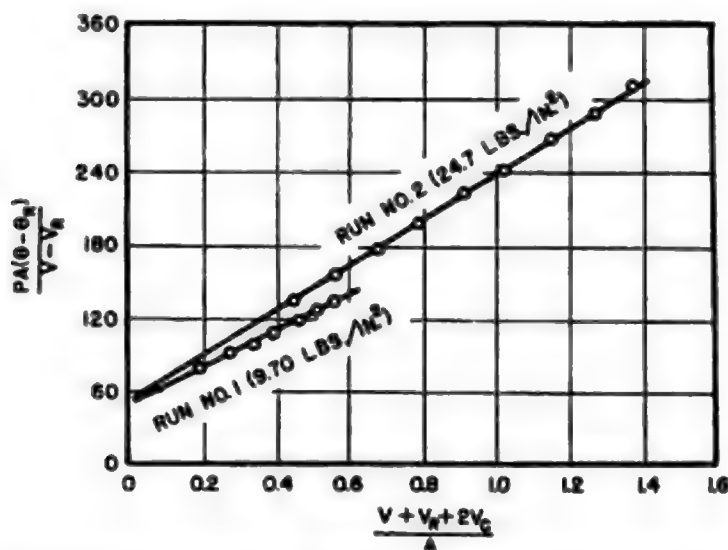


Figure 2. Experimental Results for Filtration of Calcium Carbonate Slurry at Constant Pressure

and plotted against each other, and the best straight line was drawn through the points and extrapolated to $\frac{V + V_r + 2V_c}{A} = 0$, as shown in Figure 2.

As is evident from Equation (1), the slope of each curve is $\frac{r''v\mu}{2} P^s$, and its intercept at $P = 0$ is $\mu\rho'P^m$.

2. From the known values of the slurry concentration a and the viscosity μ , values of $\frac{r''v}{a} P^s$ and $\rho'P^m$ were calculated for each run, and these values, which are given in Table 4, were then plotted against P on log-log paper in Figures 3 and 4.

The curve of $\frac{r''v}{a} P^s$ versus P has a slope of s and an intercept at

TABLE 4

Comparison of Experimentally Determined Filtration Constants for CaCO_3 with Values Calculated from Published Data^{1,2}

Item	From published data			From experimental results	
	10	15	20	9.70	24.7
Pressure, lb./in. ²	10	15	20	9.70	24.7
Cloth resistance, $\rho'P^m$	61.2	17.1	48.5	52	56
Cake resistance, $\frac{r''v}{a}P^s$	1107	1195	785	322	378
Cake resistivity exponent, s	0.14 and 0.09			0.17	
Cloth resistivity exponent, m	—			0.088	
Cloth resistivity, ρ'	—			43	

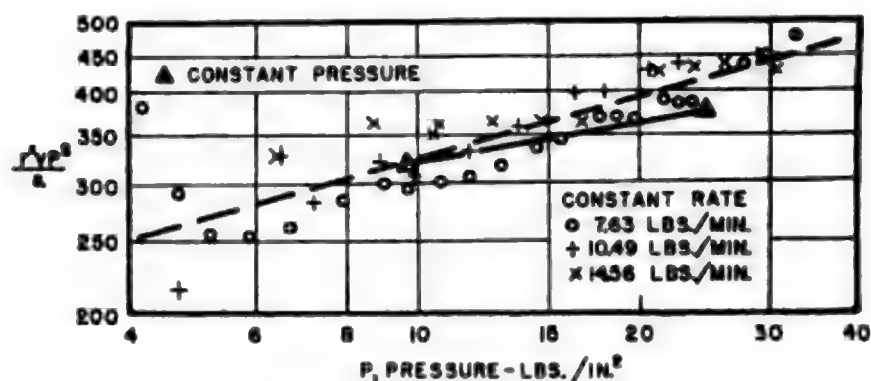


Figure 3. Variation of Cake Resistance of Calcium Carbonate with Pressure

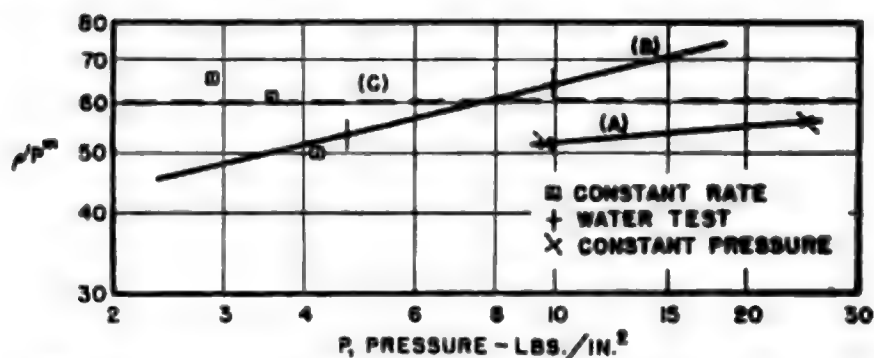


Figure 4. Experimental Results for Filter Cloth Resistance

$P = 1$ of $\frac{r''v}{a}$, and the curve of $\rho'P^m$ versus P has a slope of m and an intercept at $P = 1$ of ρ' . The values of these quantities were found to be: $s = 0.17$, $m = 0.088$, $\rho' = 43$, and $\frac{r''v}{a} = 217$.

3. From the data of the water test for the determination of the cloth resistance, values of $\rho'P^m$ were calculated from the equation

$$\frac{PA\theta}{V\mu} = \rho'P^m \quad (2)$$

and plotted against P on log-log paper as Curve *B* of Figure 4. The slope of this curve is m and its intercept at $P = 1$ is ρ' ; the values were found to be 0.46 and 37 respectively.

Sample Calculations. The various calculations were made as follows:

1. Values of $\frac{P(\theta - \theta_r)A}{V - V_r}$ and $\frac{V + V_r + 2V_e}{A}$. For example, the data for Reading No. 3 of Filtration Run No. 1 (Table 2) are:

$$P = \text{gage pressure} + \text{correction} = 10 - 0.3 = 9.7 \text{ lb./in.}^2$$

$$\theta = 1 \text{ min. } 10 \text{ sec.} = 70/60 \text{ min.}$$

$$\theta_r = 13 \text{ sec.} = 13/60 \text{ min.}$$

$$A = 4 \times \text{area of one frame-side} = 4 \times 85.6 = 342.4 \text{ in.}^2$$

$$\mu = 1.0 \text{ centipoises at } 20.5^\circ\text{C.}$$

$$V = 47.75 \text{ lb.}$$

$$V_r = 7.75 \text{ lb.}$$

$$V_e = 2.5 \text{ lb.}$$

$$a = 1 \text{ lb./100 lb. filtrate.}$$

Therefore,

$$\frac{P(\theta - \theta_r)A}{V - V_r} = \frac{9.7(70 - 13) \times 342.4}{(47.75 - 7.75) \times 60} = 78.9$$

and

$$\frac{V + V_r + 2V_e}{A} = \frac{47.75 + 7.75 + 2 \times 2.5}{342.4} = 0.177$$

The other points used in plotting the curves of Figure 2 were calculated in a similar manner.

2. *Cake and Cloth Resistances from the Slurry Test Data.* For example, for Run No. 1, the slope of the curve (Figure 2) is 161 and the intercept is 52.

Since the slope is $\frac{r''v\mu}{2} P^s$,

$$\frac{r''vP^s}{a} = \frac{r''v\mu}{2} P^s \times \frac{2}{\mu a} = 161 \times \frac{2}{1.0 \times 1} = 322$$

and since the intercept is $\mu\rho'P^m$,

$$\rho'P^m = \frac{\text{intercept}}{\mu} = \frac{52}{1.0} = 52$$

For Run No. 2, the corresponding values of $\frac{r''v}{a}P^s$ and $\rho'P^m$ were found to be 378 and 56, respectively.

3. *Cloth Resistance from the Water Test Data.* For example, the average results of Runs No. 1 and 2 (Table 3) are:

$$P = \text{gage pressure} + \text{correction} = 5 - 0.3 = 4.7 \text{ lb./in.}^2$$

$$\theta = 38.5/60 \text{ min.}$$

$$A = 2 \times 85.6 = 171.2 \text{ in.}^2$$

$$V = 10 \text{ lb.}$$

$$\mu = 0.96 \text{ centipoises at } 22^\circ\text{C.}$$

Substituting these values in Equation (2):

$$\begin{aligned}\rho'P^m &= \frac{PA\theta}{V\mu} = \frac{4.7 \times 171.2 \times 38.5}{10 \times 0.96 \times 60} \\ &= 53.8\end{aligned}$$

The corresponding value for the average of Runs No. 3 and 4 is 62.4

Comparison of Results with Published Data. Values of cloth and cake resistances, $\rho'P^m$ and $\frac{r''v}{a}P^s$, for calcium carbonate calculated from published data⁴ and obtained from the present experimental work are given in Table 4. Although no significant comparison can be made of these results because no information is given as to the type of cloth used in the other investigations and the method of preparing the slurries, it is evident that the precipitated chalk used in the present experimental work was much coarser than the CaCO_3 used in the reported studies.

Values of the cake resistivity exponent s for CaCO_3 are reported in the literature⁵ as 0.14 and 0.09; these values are somewhat lower than the value of 0.17 obtained from the two constant-pressure runs in the present work.

TYPICAL CONSTANT-RATE FILTRATION EXPERIMENT

Objects. To study the filtration characteristics of a slurry of precipitated chalk at constant rate, and to determine the resistance of the filter cloth.

Procedure. 1. Assemble the filter press with a single frame, and

then install at *Q* an orifice plate that will give a differential head of 4 to 12 inches of water under air at the maximum contemplated flow rate.

2. With by-pass valve *L* closed, fill the press and filtrate line with water through valve *K*, and then flush out all air from the manometer lines by opening cocks *S* and *T* and pinch clamp *U*. Close cocks *S* and *T* and open cock *V* until sufficient air has been drawn into the manometer through *U* to give a suitable level, and then close pinch clamp *U* and cock *V* and drain the press. *Do not* open cocks *S* and *T* until the filtrate starts to issue during a run, and be sure to close them at the end of the run before the press is opened.

3. Make 3 runs at different rates using a slurry prepared from about 250 gallons of water and 1 to 2 per cent of precipitated chalk. In making each run, open valve *L*, start the pump, and then immediately adjust valve *L* to obtain the desired manometer reading; continue to adjust this valve during the run so that the reading remains constant.

4. Record the time, pressure, filtrate weight, and manometer readings at regular intervals, such as every 10 pounds, and check the constancy of the flow rate from time to time from the weights of filtrate.

5. Determine the pressure correction by measuring the difference in elevation between the center of the gage and the filtrate discharge.

Analysis of the Data. Constant-rate filtration data are analyzed by means of the equation³

$$\frac{dV}{d\theta} = \frac{PA^2}{\mu[r''vP^s(V + V_c) + \rho'AP^m]} \quad (3)$$

where

$\frac{dV}{d\theta}$ = filtration rate, lb./min.

P = pressure differential across cloth and cake, lb./in.²

A = cloth area, in.²

μ = viscosity of filtrate, centipoises.

r'' = cake resistivity.

v = volume of cake, in.³/lb. filtrate.

s = cake resistivity exponent.

$(V + V_c)$ = weight of filtrate collected plus weight of filtrate trapped in filtrate line, lb.

ρ' = cloth resistivity.

m = cloth resistivity exponent.

1. If only one run is made, the corrected pressure readings *P* are plotted against the corrected filtrate readings $(V + V_c)$:

a. *If the curve is a straight line*, the pressure increases at a uniform rate and, therefore, both s and m are zero and both P^s and P^m are equal to 1. Equation (3) then becomes

$$P = \frac{r''v\mu}{A^2} \frac{dV}{d\theta} (V + V_c) + \frac{\rho'\mu}{A} \frac{dV}{d\theta} \quad (4)$$

which is the equation of the line. The slope of this line, therefore, is $\frac{r''v\mu}{A^2} \frac{dV}{d\theta}$ and its intercept at $(V + V_c) = 0$ is $\frac{\rho'\mu}{A} \frac{dV}{d\theta}$. From these values and the concentration of the slurry a , the quantities $\frac{r''v}{a}$ and ρ' can be calculated.

b. *If the intercept of the curve is negligible*, then ρ' is negligible and can be taken as zero. In this case, Equation (3) becomes

$$P = \frac{r''\mu P^s}{A^2} (V + V_c) \frac{dV}{d\theta} \quad (5)$$

which may be written as

$$\frac{PA^2}{\mu a(V + V_c) \frac{dV}{d\theta}} = \frac{r''v}{a} P^s \quad (6)$$

If values of $\frac{PA^2}{\mu a(V + V_c) \frac{dV}{d\theta}}$ are plotted against P on log-log paper and

the best straight line is drawn through the points, the slope of this line will be s and the intercept at $P = 1$ will be $\frac{r''v}{a}$.

c. *If the curve is not straight or if the intercept is not negligible*, the following method is used:¹

(1) The value of m is calculated by trial and error solution of the equation

$$\frac{V_2 V_4}{V_3^2} = \frac{(R - R^m)(R^3 - R^{2m})}{(R^2 - R^{2m})^2} = 1 - \frac{1}{2 + R^{1-m} + \frac{1}{R^{1-m}}} \quad (7)$$

where

$R \doteq \frac{P_4}{P_3} = \frac{P_3}{P_2} = \frac{P_2}{P_1}$, and the subscripts 1, 2, 3 and 4 refer to corresponding readings of V and P .

If the value of m comes out negative, a value of $m = 0$ is taken, and if m is greater than 0.5, a value of 0.5 is used.

(2) From values of m and P_* , which is P extrapolated back to $(V + V_c) = 0$, the value of ρ' is calculated from Equation (3) which at $(V + V_c) = 0$ becomes

$$\frac{P_* A}{\mu dV/d\theta} = \rho' P_*^m \quad (8)$$

(3) Equation (3) may be rewritten as

$$\frac{r''v}{a} P^s = \frac{A}{a(V + V_c)} \left(\frac{PA}{\mu \frac{dV}{d\theta}} - \rho' P^m \right) \quad (9)$$

and used to calculate values of $\frac{r''v}{a} P^s$, which are then plotted against P on log-log paper; the slope of the line is s and the intercept at $P = 1$ is $\frac{r''v}{a}$.

An alternate procedure is to determine ρ' and m with water alone, as described in the constant pressure experiment, and to use these values to determine s and $\frac{r''v}{a}$ by means of Equations (8) and (9).

2. When data from several runs are available, the following method is used:

a. For each run, values of $(V + V_c)$ are plotted against P and the curves are extrapolated to $(V + V_c) = 0$, and for each value of the intercept P_* , values of the cloth resistance $\rho' P_*^m$ at $(V + V_c) = 0$ are calculated by means of Equation (8) and plotted against P_* on log-log paper, and the best straight line is drawn through the points. As shown by this equation, the slope of the line is m and the intercept at $P_* = 1$ is ρ' .

b. For each run, values of the cake resistance $\frac{r''v}{a} P^s$ are calculated from Equation (9) and plotted against P on log-log paper. As shown by this equation, the best straight line through these points has a slope of s and an intercept at $P = 1$ of $\frac{r''v}{a}$.

Experimental Data and Results. Experimental data for three constant rate runs are given in Table 5. These data were analyzed by the method outlined under 2 above:

TABLE 5
Experimental Data for Filtration of CaCO_3 at Constant Rate

Run No. 1				Run No. 2			Run No. 3		
Reading No.	Time θ , min.	Pressure gage reading, lb./in. ²	Filtrate collected V , lb.	Time θ , min.	Pressure gage reading, lb./in. ²	Filtrate collected V , lb.	Time θ , min.	Pressure gage reading, lb./in. ²	Filtrate collected V , lb.
1	1.33	4.5	10	1.33	5.0	10	0.83	7.0	10
2	2.66	5.0	20	2.28	6.8	20	1.52	9.3	20
3	3.92	5.5	30	3.25	7.5	30	2.20	11.0	30
4	5.25	6.2	40	4.33	9.2	40	2.92	13.0	40
5	6.53	7.0	50	5.01	10.2	50	3.57	15.0	50
6	8.92	8.2	60	6.03	12.0	60	4.25	17.0	60
7	9.17	9.3	70	7.00	14.0	70	5.00	21.5	70
8	10.17	10.0	80	7.92	16.7	80	5.67	24.0	80
9	11.75	11.0	90	8.92	18.3	90	6.33	26.5	90
10	13.00	12.0	100	9.92	21.0	100	7.00	29.5	100
11	14.33	13.2	110	10.87	23.0	110	7.70	31.0	110
12	15.75	14.8	120						
13	17.00	16.0	130						
14	18.25	18.0	140						
15	20.58	18.9	150						
16	21.00	20.0	160						
17	22.25	22.0	170						
18	23.42	23.0	180						
19	24.83	24.1	190						
20	26.08	28.0	200						
21	27.50	30.0	210						
22	30.58	33.0	220						

Item	Run No. 1	Run No. 2	Run No. 3
Filtrate correction V_c , lb.....	2	2	2
Pressure correction, lb./in. ²	-0.3	-0.3	-0.3
Filtering area, in. ²	171.2	171.2	171.2
Slurry concentration a , lb. solids/100 lb. filtrate...	2.18	2.18	2.18
Lbs. dry cake/100 lbs. filtrate.....	2.11	2.27	2.38
No. of frames.....	1	1	1
Temperature of filtrate, °F.....	69.2	69.2	69.0
Manometer reading, in. water.....	2	4	8
Calculated flow rate, lb./min.....	7.63	10.49	14.56

1. The curves obtained by plotting $(V + V_c)$ versus P for each run are shown in Figure 5. These curves were extrapolated to $(V + V_c) = 0$, and values of P_s were read as follows:

Run No.	1	2	3
P_s , lb./in. ²	2.9	3.6	4.2

For each of these intercept readings, values of $\rho'P^m$ were calculated by means of Equation (8) and plotted against P , in Figure 4.

2. For each run, values of $\frac{r''v}{a}P^s$ were calculated by means of Equation (9) and plotted against P in Figure 3. The best curve was drawn through all of the points for the three runs: the slope s and the intercept $\frac{r''v}{a}$ were found to be 0.27 and 171, respectively.

Summary or Results. 1. The cake and cloth resistances for the constant rate runs, as shown in Figures 3 and 4, check fairly closely

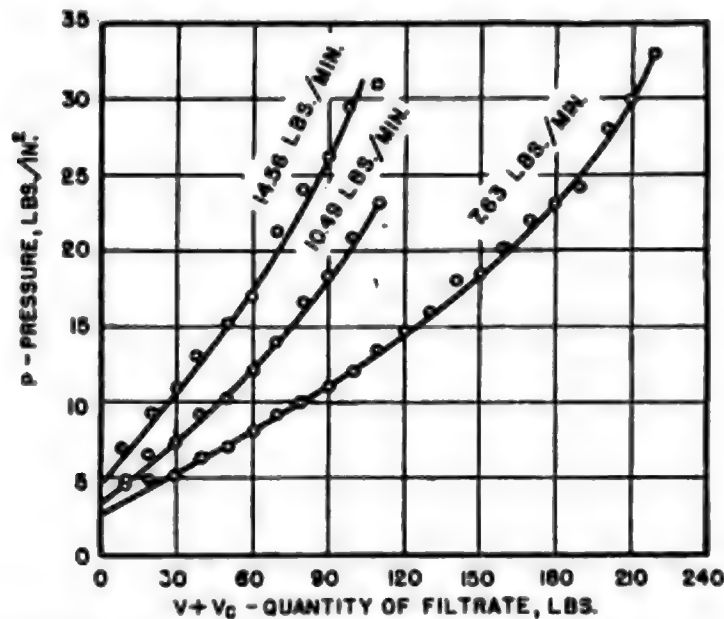


Figure 5. Experimental Results for the Filtration of Calcium Carbonate at Constant Rate

with the values obtained at constant pressure.² Curve *C*, of Figure 4, which is based on the results of the constant rate, the constant pressure, and the water test experiments, indicates that over a pressure range of about 0 to 30 pounds per square inch and for a 1 to 2 per cent slurry of precipitated chalk, the cloth resistivity ρ' is 60 and the cloth resistance exponent m is practically 0.

2. The curves of Figure 3 show that the cake resistivity exponent s is somewhat higher at constant rate than at constant pressure; this is usually the case.

NOMENCLATURE

a = concentration of slurry, lb. solid/100 lb. filtrate.

A = cloth area, in.²

m = cloth resistivity exponent.

P = pressure differential across cloth and cake, lb./in.²

P_e = value of P extrapolated back to $(V + V_e) = 0$.

r'' = cake resistivity.

$\frac{r''v}{a} P^s$ = cake resistance.

s = cake resistivity exponent.

v = volume of cake, in.³/lb. filtrate.

V = weight of total filtrate collected, lb.

V_e = weight of filtrate trapped in filtrate line, lb.

V_r = weight of filtrate collected during θ_r , lb.

θ = total time of filtering, min.

θ_r = time of filtering before desired constant pressure is reached, min.

μ = viscosity of filtrate, centipoises.

ρ' = cloth resistivity.

$\rho' P^m$ = cloth resistance.

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3. WALKER, LEWIS, McADAMS, AND GILLILAND, "Principles of Chemical Engineering," McGraw-Hill Book Company, 1937, 3rd ed., pp. 346-7.
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A SETUP FOR VACUUM FILTRATION AND TESTING OF FILTER AIDS

Designed and Constructed by

The Department of Chemical and Gas Engineering
Johns Hopkins University*

Description. This setup, which is shown in Figure 1, includes: (1) a 10-inch Sweetland filter leaf (Figure 2) connected to two filtrate receivers made of inverted gas cylinders maintained under vacuum, and (2) a 52-gallon steel drum provided with air agitation. A mercury-under-water manometer is connected between the filtrate line and the drum to indicate the filtration pressure, and water-level gages are provided on the cylinders to indicate the quantity of filtrate collected during an experiment. The setup is particularly suited to the study of filter aids because these erosive materials do not have to pass through any pump.

Materials and Cost. The materials necessary for this setup, as illustrated, are listed in Table 1. The cost of these materials, exclusive of the gas cylinders, is about \$50.

Auxiliary Equipment. The following auxiliary equipment is needed when making a run with this setup:

1. Vacuum pump: about 10 cubic feet per minute, and preferably capable of providing 29 inches of vacuum. ✓
2. Cycloidal type of blower: about 10 cubic feet per minute against 5 inches of mercury.
3. Thermometer: 0 to 100°C. in 1°C. or better.
4. Stop watch: 60 seconds in $\frac{1}{2}$ second for $\frac{1}{2}$ hour.
5. Scale to weigh charge: 5 pounds capacity or greater.
6. Spatula, tin pans, drying oven, and balance (1 kilogram).

Notes on Construction. 1. For accuracy in measuring the quantity of filtrate, the receivers should have a constant cross-sectional area. If the lower portion does not have the same cross-sectional area as the rest of the cylinder, it will facilitate computations to place the zero mark of the filtrate-level gages above this portion.

2. The air for agitation of the slurry should be supplied by a positive rotary blower that operates without oil; for example, a cycloidal blower.

3. If old gas cylinders are not available, standard 6- or 8-inch pipes provided with caps or welded ends may be used as filtrate receivers.

* Reported by C. F. Bonilla.

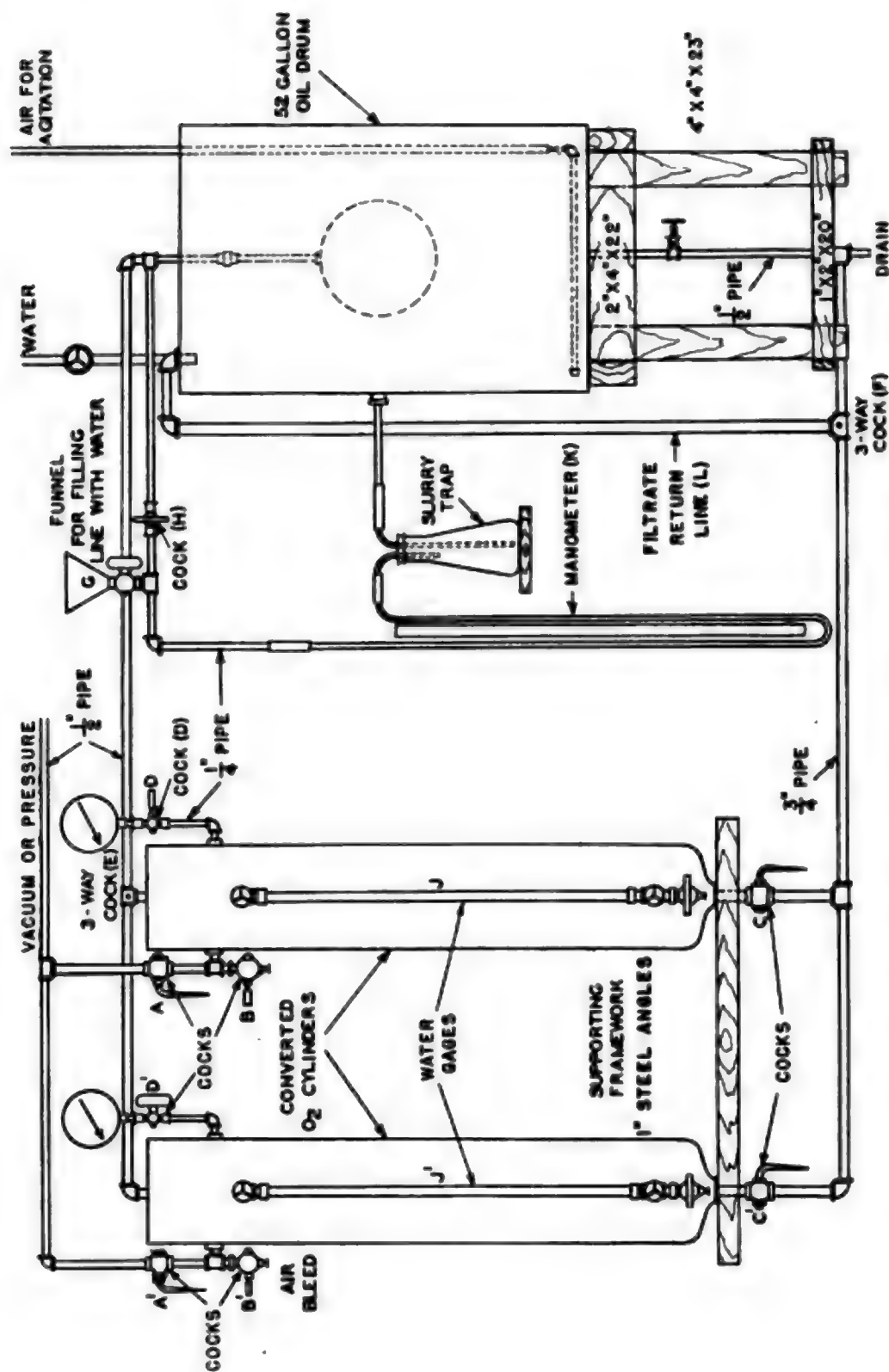


Figure 1. Setup for Vacuum Filtration with a Single Leaf—Johns Hopkins University

4. The drum should be painted on both the inside and outside to prevent rusting.

Comments on Design. 1. The setup has a provision for connecting the vacuum pump to the vacuum line in a reversed position so that the filtrate that accumulates in the two receivers can be blown back into the slurry tank. Additional runs can then be made with the same materials, provided the cake can be thoroughly dispersed. However, if it is not desired to return the filtrate for repeat runs, three-way cock *F* and filtrate-return line *L* may be eliminated.

2. Funnel *G*, which is used to fill the manometer lead with water, should preferably be closer to the mercury than is shown in the drawing.

3. Mild, mechanical agitation may be used in place of air agitation.

4. Although a 10-inch filter leaf was used with this setup, nominal 8½- and 16-inch leaves are also available.

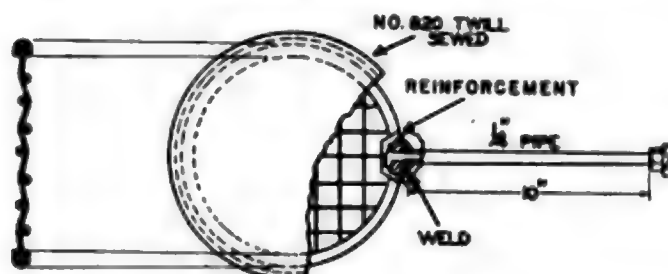


Figure 2. Details of Single Filter Leaf

5. For slurries which offer considerable resistance, one receiver of the size indicated may be sufficient to collect all of the filtrate during a run.

6. The filter cake builds up to a considerable thickness over the entire cloth, even over and beyond the rim. In order to accurately fix the filtration area, it is desirable to either paint or otherwise blank off the cloth over and beyond the rim or else to drill a number of small holes through the rim to make that area available for filtration.

Operation. 1. Water, which is run into the receivers, or filtrate which is left from a previous test, is run out to the desired level, preferably the zero mark on gages *J* and *J'*. Cocks *C* and *C'* are then closed.

2. A slurry is made up in the drum with thorough agitation.

3. Water is added through funnel *G* until all of the air is displaced from the manometer line. The funnel and cock *H* are then closed, and the filter leaf is connected to the filtrate line by means of a union.

4. Three-way cock *E* is closed, the vacuum pump is started, and the filtrate receivers are evacuated to the desired pressure, or slightly less, as indicated by the vacuum gages. This vacuum is maintained, preferably, by slightly opening cocks *A* and *B* and *A'* and *B'*.

TABLE 1
Bill of Materials for a Vacuum Filtration Setup
 The Johns Hopkins University

Quantity	Description	Size	Material
10 ft.	Std. pipe	$\frac{1}{2}$ in.	Galv. wrought steel
20 ft.	" "	$\frac{1}{2}$ in.	" " "
10 ft.	" "	$\frac{1}{2}$ in.	" " "
4	Std. nipple	$\frac{1}{2}$ in. (assorted lengths)	" " "
11	" "	$\frac{1}{2}$ in. (assorted lengths)	" " "
7	" "	$\frac{1}{2}$ in. (assorted lengths)	" " "
2	Std. tee	$\frac{1}{2}$ in.	Galv. malleable iron
3	" "	$\frac{1}{2}$ in.	" " "
1	" "	$\frac{1}{2}$ in.	" " "
2	Std. 90° elbow	$\frac{1}{2}$ in.	" " "
6	" " "	$\frac{1}{2}$ in.	" " "
4	" " "	$\frac{1}{2}$ in.	" " "
1	Std. side-outlet elbow	$\frac{1}{2}$ in.	" " "
1	Ground-joint union	$\frac{1}{2}$ in.	" " "
3	" " "	$\frac{1}{2}$ in.	" " "
1	" " "	$\frac{1}{2}$ in.	" " "
2	Bushing	$\frac{1}{2} \times \frac{1}{2}$ in.	" " "
1	Plug	$\frac{1}{2}$ in.	" " "
5	Std. globe valve	$\frac{1}{2}$ in.	Brass
2	Lever-handle cock	$\frac{1}{2}$ in.	"
2	" " "	$\frac{1}{2}$ in.	"
1	" " "	$\frac{1}{2}$ in.	"
1	Three-way cock (3 port)	$\frac{1}{2}$ in.	"
1	Three-way cock (3 port)	$\frac{1}{2}$ in.	"
2	Pet cock	$\frac{1}{2}$ in.	"
3	Gage cock	$\frac{1}{2}$ in.	"
1	Water gage, low pressure	$\frac{1}{2}$ in., with 36 in. glass	"
2	Vacuum gage	5 in. dial	"
1	Manometer	$\frac{1}{2}$ in. O.D. x 36 in.	Glass
2	Gas cylinder	1.3 cu. ft. capacity	Steel
1	Sweetland filter leaf	10 in.	Bronze
2 sq. ft.	Filter cloth	16.5 oz./sq. yd.	Cotton
1	Drum	55 gal.	Steel
18 ft.	Angles	1 x 1 x $\frac{1}{2}$ in.	"
8 ft.	Lumber	2 x 4 in.	Pine
20 ft.	"	1 x 2 in.	"
8 ft.	"	4 x 4 in.	"

5. The stop watch is started, and cock *E* is opened simultaneously to connect one of the cylinders. Cock *H* is then opened slowly to connect the manometer to the system.

6. From this point on, *A* and *B* or *A'* and *B'* are adjusted continuously to maintain the reading on manometer *K* constant throughout the run.

7. The filtrate level is read and recorded at regular intervals, for example every $\frac{1}{4}$ minute.

8. If the quantity of filtrate is large, it can be run into the second cylinder by turning cock *E*; the vacuum to the filled cylinder may then be broken, the filtrate may be drained to the sewer, and the receiver may be re-evacuated and used again if needed.

9. At the end of the run, the cake is scraped into a tared pan and weighed, and then dried and reweighed. From these weighings, the quantity of solids in the slurry and the moisture content of the cake are determined.

TYPICAL EXPERIMENT

Object. To study the vacuum filtration characteristics of a slurry of precipitated aluminum hydroxide, with and without filter aids.

Procedure. 1. Prepare an aqueous slurry of about 0.1 per cent of aluminum hydroxide by weight by dissolving potassium alum, $\text{Al}_2(\text{SO}_4)_3 \cdot \text{K}_2\text{SO}_4 \cdot 24\text{H}_2\text{O}$, in the tank and adding a solution of soda ash, Na_2CO_3 , until the slurry is neutral to litmus.

2. Calculate the hold-up in the filtrate line. For simplicity in computing the amount of filtrate, adjust the water level in the first receiver so that it contains a quantity above the zero mark of the scale equivalent to the computed hold-up.

3. Carry out two duplicate runs or two runs at considerably different pressures.

4. Repeat the experiment after adding a weight of Hyflo supercel filter aid equal to about 50 per cent of the remaining $\text{Al}(\text{OH})_3$.

5. Repeat the experiment again after adding another 50 per cent of the filter aid to the remaining $\text{Al}(\text{OH})_3$.

6. Analyze the data by means of the constant pressure filtration equation.

Analysis of Data. In this experiment, there was no time lag before constant pressure was reached and, therefore, both V_r and θ_r were equal to zero. In this case the data can be analyzed by the integrated rate equation (see derivation on pages 433-434):

$$\frac{P\theta A}{V + V_c} = \frac{r'v}{a} P^s \frac{\mu a(V + V_c)}{2A} + \mu\rho'P^m \quad (1)$$

where

P = pressure differential across cloth and cake, lb./in.²

θ = total time of filtering, min.

A = cloth area, in.²

V = total weight of filtrate collected, lb.

V_c = weight of filtrate trapped in filtrate line, lb.

r'' = cake resistivity.

v = volume of cake, in.³/lb. filtrate.

a = concentration of the slurry, lb. solid/100 lb. filtrate.

s = cake resistivity exponent.

μ = viscosity of filtrate, centipoises.

ρ' = cloth resistivity.

m = cloth resistivity exponent.

If b equals the pounds of filtrate per unit on the receiver gage scale, and H is the reading on the scale corrected to $H = 0$ at $V + V_c = 0$, Equation (1) can be written as:

$$\frac{P\theta A}{bH} = \frac{r''v}{a} P^s \frac{\mu abH}{2A} + \mu \rho' P^m \quad (2)$$

or as

$$\frac{\theta}{H} = \frac{r''v}{a} P^s \frac{\mu ab^2}{2A^2 P} H + \frac{\mu \rho' P^m b}{PA} \quad (3)$$

Therefore, if values of θ/H are plotted against H , the slope of the line will be $\frac{r''v}{a} P^s \frac{\mu ab^2}{2A^2 P}$ and the intercept will be $\frac{\mu \rho' P^m b}{PA}$ or

$$\frac{r''v}{a} P^s = \text{slope} \times \frac{2PA^2}{\mu ab^2} \quad (4)$$

and

$$\rho' P^m = \text{intercept} \times \frac{PA}{\mu b} \quad (5)$$

If runs at more than one pressure are made, values of s , r'' , m , and ρ' can be calculated in the following manner:

1. *Calculation of s and r'' .* Values of $\frac{r''v}{a} P^s$ are calculated by means of Equation (4) and plotted against P on log-log paper, and the best straight line is drawn through the points. This line has the equation

$$\log \frac{r''v}{a} P^s = s \log P + \log \frac{r''v}{a} \quad (6)$$

TABLE 2

Experimental Data for the Vacuum Filtration of $Al(OH)_3$ and Filter Aid

Filtrate hold-up* V_0 , lb.	= 0.7
Factor b , lb. filtrate/cm. gage scale.	= 0.82
Filter leaf** area, in. ²	= 134
Filtration temperature, °F.	= 68
Viscosity of water at 68°F., centipoises.	= 1.01
Calculated slurry*** concentration a , lb. $Al(OH)_3$ /100 lb. H_2O .	= 0.0985

Item	Run Number					
	1	2	3	4	5	6
Water in cake, wet basis, %	83.6	84.1	74.9	75.1	72.7	75.4
Approximate cake thickness, mm.	1.0	1.0	2.0	2.0	3.5	3.5
Filter aid (Hyflo supercell), %	0	0	50	50	100	100
Manometer reading, cm. Hg.	78.5	78.5	78.5	78.5	76.0	78.5
Gage reading, lb./in. ²	14.06	14.06	14.06	14.06	13.61	14.06
Time θ , min.	Filtrate level, cm. ($H = 0$ at $V + V_0 = 0$)					
0	(0.8)	(0.8)	(0.8)	(0.8)	(0.8)	(0.8)
0.25	7.5	6.9	4.6	4.3	4.5	6.5
0.50	11.8	10.7	7.0	6.8	7.2	6.8
0.75	14.6	12.8	8.9	8.7	9.5	11.9
1.00	16.7	14.6	10.6	10.3	12.0	14.0
1.50	19.6	17.7	13.2	13.1	16.0	17.9
2.00	22.1	19.6	15.6	15.3	19.0	21.6
2.50	23.9	21.4	17.7	17.4	22.0	24.7
3.00	25.7	23.1	19.5	19.3	24.7	27.4
3.50	27.3	24.5	21.3	21.1	27.3	30.3
4.00	28.9	25.9	23.1	22.6	29.5	32.2
4.50	30.2	27.2	24.5	24.3	31.6	34.5
5.00	31.4	28.5	26.2	25.8	33.8	36.9
5.50	32.7	29.6	27.5	27.1	35.5	39.0
6.00	33.9	30.9	28.9	28.5	37.8	41.0
6.50	35.0	31.8	30.2	29.8	39.5	42.9
7.00	36.1	33.0	31.5	31.1	41.2	44.7
7.50	—	33.9	32.6	32.3	42.5	46.6
8.00	—	34.9	33.8	33.5	44.5	48.1
8.50	—	—	34.8	34.7	46.0	50.2
9.00	—	—	36.2	—	47.5	52.0
9.50	—	—	37.4	—	49.0	53.3
10.00	—	—	—	—	51.0	55.0
10.50	—	—	39.3	—	52.2	56.4
11.00	—	—	40.4	—	53.6	58.0
11.50	—	—	—	—	55.0	—

* Calculated from volume of filtrate line—38 in. of $\frac{1}{2}$ -in. pipe + 24 in. of $\frac{1}{4}$ -in. pipe—and volume of filter leaf—area $\times \frac{3}{4}$ -in. thickness.

** The filter leaf is 7.75 in. I.D. and 9.50 in. O.D., but from the appearance of the cake, the average diam. was 9.25 in., therefore, area = $2\pi(9.25)^2/4 = 134$ in.².

*** The slurry was made from 40 gal. of water and 2 lb. of potassium alum.

and, therefore, its slope is s and its intercept at $P = 1$ is $\frac{r''v}{a}$, from which r'' can be calculated since v and a are both constant.

2. *Calculation of m and ρ' .* Values of $\rho'P^m$ are calculated by means of Equation (5) and plotted against P on log-log paper. The best straight line through these points has the equation

$$\log \rho'P^m = m \log P + \log \rho' \quad (7)$$

and its slope is m and its intercept at $P = 1$ is ρ' .

Experimental Data. The experimental data for six filtration runs are given in Table 2. Values of θ/H were plotted against H in Figure 3, and

TABLE 3
Calculated Results for the Vacuum Filtration of $Al(OH)_3$ and Filter Aid
(From curves of Figure 3)

Item	Run Number					
	1	2	3	4	5	6
Filter aid, %.....	0	0	50	50	100	100
Slope.....	0.00743	0.00835	0.00611	0.00616	0.00296	0.00278
Intercept.....	0.026	0.026	0.030	0.034	0.048	0.032
Cake resistance, $\frac{r''v}{a} P^s$	56,000	62,800	46,000	46,500	21,600	21,000
Cloth resistance, $\rho'P^m$	59	59	68	77	106	73

the curves were extrapolated to $H = 0$. From the slopes and intercepts, the values of $\frac{r''v}{a} P^s$ and $\rho'P^m$ were calculated by means of Equations (4) and (5); the results are summarized in Table 3.

Discussion of Results and Comparison with Published Data. 1. It is evident from Figure 3 that the data from the runs made without filter aid did not give straight line plots during the early periods of the runs. This must have been due to leakage of solids through the cloth; since this is not a constant error, no attempt was made to correct for it mathematically. However, as the points during the latter periods of the runs fell on straight lines, it was assumed that this error eventually became negligible; the cake resistances, therefore, were calculated from the slopes of the straight portions.

2. Values of s and m could not be determined for these experiments because the pressure was not varied appreciably. Values of 0.56 to

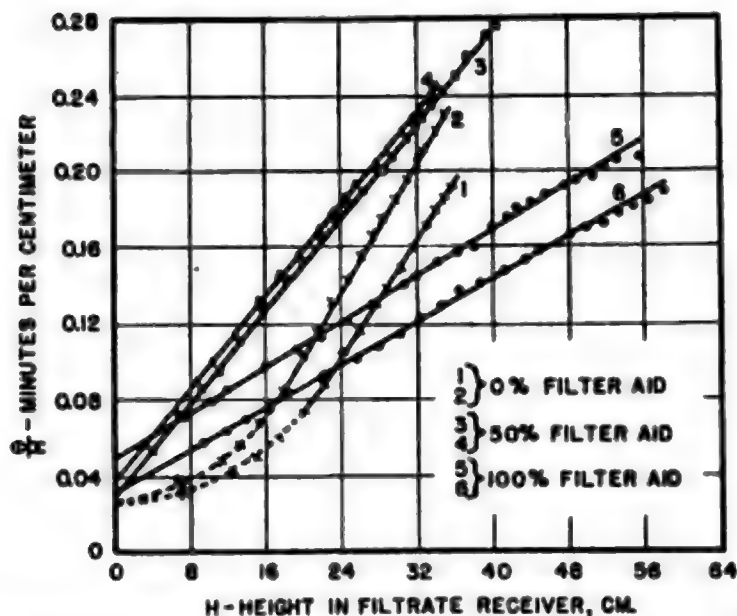


Figure 3. Experimental Results of Vacuum Filtration of Aluminum Hydroxide With and Without Filter Aid

TABLE 4

Comparison of the Filtration Constants for $Al(OH)_3$ with and without Filter Aid

Pressure, lb./in. ²	Per cent Kieselguhr, based on $Al(OH)_3$	Cake resistance, $\frac{r'v}{a} P^s$	Cake resistance compared to zero per cent filter aid	Reference
10	0	118,200	—	(3)
15	0	80,300	—	(3)
22	0	106,300	—	(3)
40	0	190,000	—	(3)
24.5	unstated amount	36,900	—	(3)
25	0	145,000	1.000	(1)*
25	50	89,000	0.613	(1)*
25	100	59,000	0.406	(1)*
14	0	59,400 (av.)	1.000	This test
14	50	46,250 (av.)	0.778	This test
14	100	21,300 (av.)	0.358	This test

* Carman's results were converted to the units used here by the relationship $\frac{r'v}{a} P^s = r_1/13,000$.

0.88 for the cake resistivity exponent s for $Al(OH)_3$ have been reported in the literature;³ these values no doubt would have been less if filter aid had been used.

3. The values of the cloth resistance $\rho'P^m$ obtained in the experiments (Table 3) check fairly well with the average value of 60 obtained with

the same cloth in a filter press with a CaCO_3 slurry (see Figure 4, page 446). The much higher value obtained in Run No. 5 is probably due to some constant error, possibly in time θ .

4. Values of the cake resistance $\frac{r''v}{a} P^s$ for $\text{Al}(\text{OH})_3$ at several different pressures calculated from data in the literature and from the results of the present experiment are given in Table 4.

On the whole, the results of the present investigation are in fair agreement with those of previous investigations, in spite of differences in methods of preparing the slurries.

NOMENCLATURE

- a = concentration of slurry, lb. solid/100 lb. filtrate.
- A = cloth area, in.²
- b = lb. filtrate per cm. change in level of filtrate in receiver.
- H = level in filtrate receiver, including correction for quantity retained in filtrate line, cm.
- m = cloth resistivity exponent.
- P = pressure differential across cloth and cake, lb./in.²
- P_0 = value of P extrapolated back to $(V + V_c) = 0$.
- r'' = cake resistivity.
- $\frac{r''v}{a} P^s$ = cake resistance.
- s = cake resistivity exponent.
- v = volume of cake, in.³/lb. filtrate.
- V = total weight of filtrate collected, lb.
- V_c = weight of filtrate trapped in filtrate line, lb.
- V_r = weight of filtrate collected during θ_r , lb.
- θ = total time of filtering, min.
- θ_r = time of filtering before desired constant pressure is reached min.
- μ = viscosity of filtrate, centipoises.
- ρ' = cloth resistivity.
- $\rho' P^m$ = cloth resistance.

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A SMALL LEAF-TYPE PRESSURE FILTER

Designed and Constructed by
The Agricultural By-Products Laboratory
Ames, Iowa

Description. This filter, described by Clark, Porges, and Aronovsky,¹ is of simple design and construction. It has three square filter leaves, arranged for either top or bottom drainage, which provide an effective filtering surface of 54 square inches. The wet cake capacity of the filter is 42 cubic inches. The filter, constructed as shown in Figures 1, 2 and 3, has been used successfully at pressures up to 30 pounds per square inch gage.

Materials and Cost. With the exception of the stainless-steel machine screws and the external tie rods, the filter is fabricated from commercial aluminum. The total cost of the materials, which are listed in Table 1, is approximately \$15.

Notes on Construction. *The Filter Case.* The filter case (Figure 1) consists of a square shell and top and bottom closing plates:

1. Shell *A* (Figure 3) is made from four $\frac{1}{4}$ -inch plates, $4\frac{1}{2} \times 5$ inches. The short sides of these plates are mitered and welded together with aluminum by means of atomic hydrogen.

2. Top and bottom closing plates *D* and *C* (Figure 3) are six inches square and $\frac{1}{2}$ -inch thick. They are held in place by eight steel tie rods secured by lock nuts. Rubber gaskets are used at the top and bottom to prevent leakage.

3. A hole is drilled in the bottom plate and tapped for a $\frac{1}{2}$ -inch pipe, which serves as the inlet port.

Filter Leaves. The filter leaves are assembled as shown in Figure 2:

1. Frames *E* (Figure 3) are made by cutting 3-inch square openings in 4-inch square plates, $\frac{3}{4}$ -inch thick; these frames are rounded to prevent cutting of the filter cloth. A hole is drilled in the top of each frame and tapped from both ends to receive $\frac{1}{8}$ -inch I.P.S. aluminum pipes through which the filtrate is discharged. When top drainage is desired, the inside tube is removed.

2. Each frame is covered with woven-aluminum filter cloth *F*, held in place by aluminum retainers, or frame covers, *G*, that are $\frac{1}{8}$ -inch thick and correspond in shape to that of the leaf frame. Stainless-steel machine screws and nuts securely hold the frame covers and filter cloths to the frames.

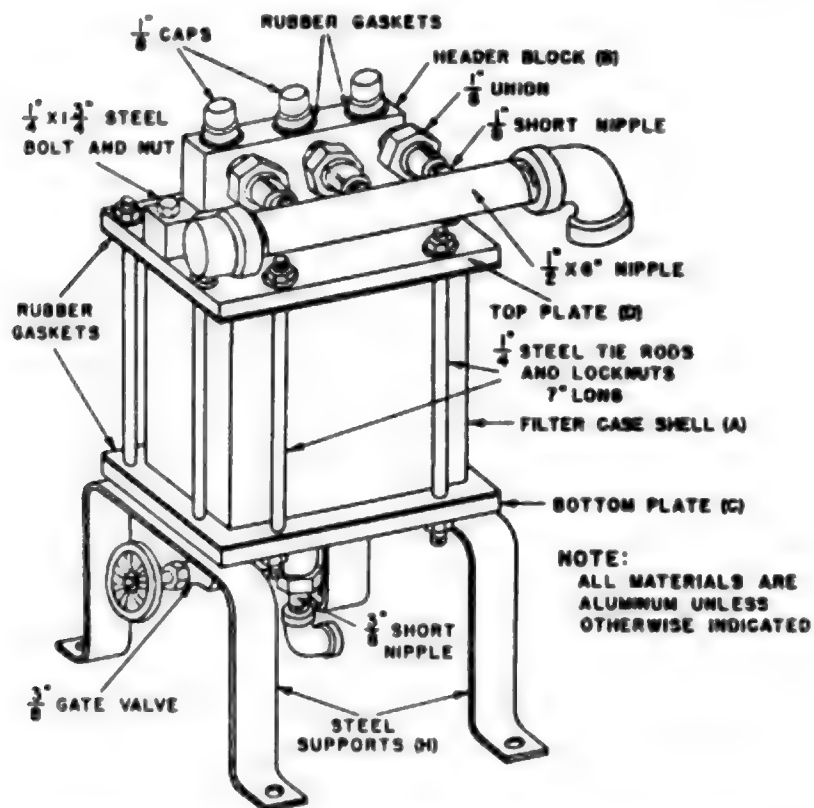


Figure 1, Assembly of Leaf-Type Pressure Filter—Agricultural By-Products Laboratory

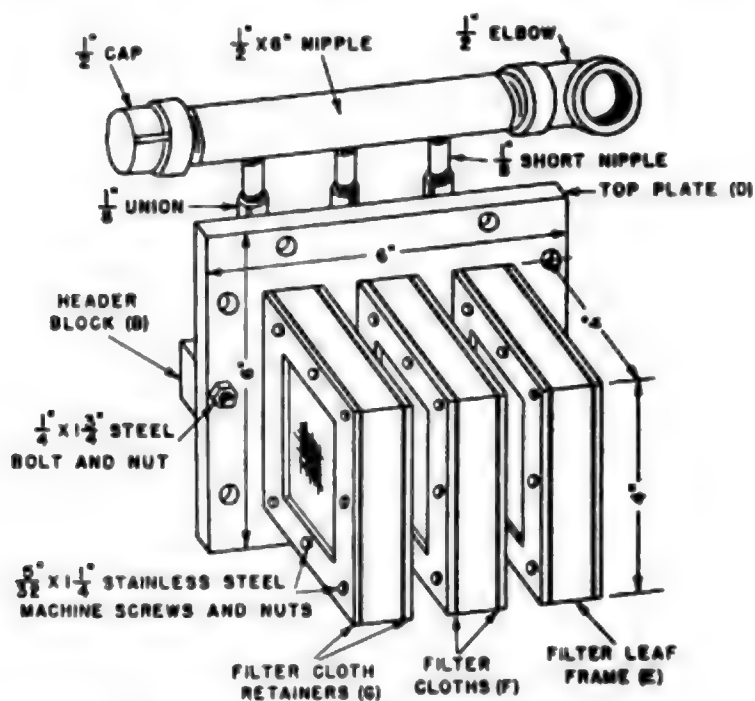


Figure 2. Assembly of Filter Leaves

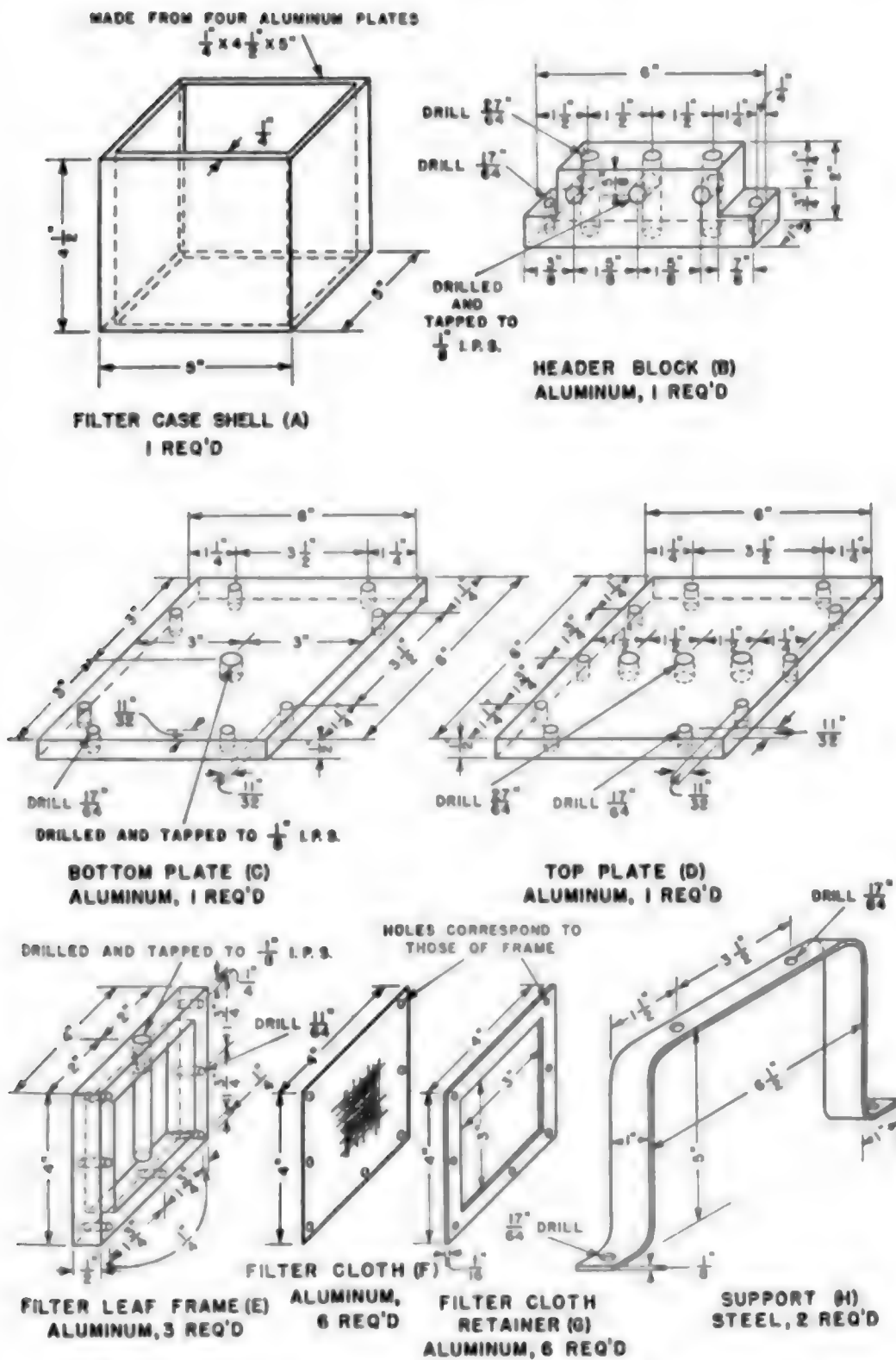


Figure 3. Details of Construction of Leaf-Type Pressure Filter

Header Block and Manifold Assembly. The filtrate is discharged through a header block and manifold assembly (Figure 1):

1. The block is constructed from a piece of aluminum bar stock, 6 x 2 x 1 inches. It is fitted with a thin rubber gasket, and bolted to the top plate as shown in Figure 1. Three holes are drilled vertically through the top plate and through the header block to accommodate the discharge tubes from the filter leaves. Three discharge ports are

TABLE 1
Bill of Materials for Leaf-type Pressure Filter
Agricultural By-Products Laboratory

Quantity	Description	Size	Material
6 in.	Pipe	$\frac{1}{2}$ in.	Aluminum
2	Nipple	$\frac{3}{8}$ in., short	"
6	"	$\frac{1}{2}$ in., "	"
1	90° elbow	$\frac{1}{2}$ in.	"
1	" "	$\frac{3}{8}$ in.	"
3	Ground-joint union	$\frac{1}{2}$ in.	"
1	Cap	$\frac{1}{2}$ in.	"
3	"	$\frac{1}{2}$ in.	"
1	Gate valve	$\frac{3}{8}$ in.	"
18 in.	Tubing	$\frac{1}{2}$ in., I.P.S.	"
1 piece	Bar stock	6 x 1 x 2 in.	"
4	Plate	$4\frac{1}{2}$ x 5 x $\frac{1}{2}$ in.	"
2	"	6 x 6 x $\frac{1}{2}$ in.	"
3	"	4 x 4 x $\frac{1}{2}$ in.	"
6	"	4 x 4 x $\frac{1}{8}$ in.	"
6	Woven cloth	4 x 4 in.	"
8	Tie rods and lock nuts	$\frac{1}{2}$ x 7 in.	Steel
32 in.	Strap	1 x $\frac{1}{2}$ in.	"
21	Machine screws and nuts	$\frac{1}{2}$ in. — 32 NS x $1\frac{1}{2}$ in.	Stainless steel
2	Bolts and nuts	$\frac{1}{2}$ in. — 20 NC x $1\frac{1}{2}$ in.	Steel

drilled in one side of the block to meet the vertical holes and these are tapped to receive $\frac{1}{8}$ -inch I.P.S. aluminum nipples to which the header is connected by means of aluminum unions. Because of the small size of this filter, the end discharge ports must be displaced sufficiently to allow clearance for the unions.

2. The discharge tubes from the top of the filter leaves extend through the vertical holes in the header block and are secured at the upper ends by $\frac{1}{8}$ -inch aluminum pipe caps. Leakage of the filtrate is prevented by means of rubber gaskets between the top plate and filter leaves and by

a combination of rubber gaskets and aluminum washers between the pipe caps and header block.

3. A $\frac{1}{4}$ -inch hole, drilled in the discharge tube at the position of the discharge port in the header block, permits the filtrate to pass from the tubes to the manifold.

4. The manifold is constructed from a section of $\frac{1}{2}$ -inch aluminum pipe. Three holes are drilled in the pipe and tapped for $\frac{1}{4}$ -inch I.P.S. in positions corresponding to those of the discharge ports in the header block. The manifold pipe is threaded at both ends: one end is provided with a cap, and the other with a 90° elbow.

Filter Supports. In order to provide space for valves and fittings beneath the filter, two supports *H* (Figure 3), formed from steel strap, are fastened to the lower ends of the tie rods.

Comments on Design. This filter was made small in size and constructed of aluminum because it was to be used in conjunction with fermentation studies where relatively small amounts of material were to be handled and where metal contamination was an important factor. The size of the filter and the material of construction, however, will depend largely upon individual requirements and the nature of the substance to be filtered.

The construction of this filter can be improved by:

1. Welding the bottom plate to the filter case shell.
2. Welding the header block to the top plate.
3. Welding a flanged section around the top of the square filter case to permit the use of short bolts instead of long tie rods.
4. Installing a vent in the top plate to permit the exhaustion of air entrapped when bottom drainage is employed.

Acknowledgement. The courtesy of Industrial and Engineering Chemistry is acknowledged for permission to use the material embodied in this report.

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SECTION 8

CRUSHING AND GRINDING

INTRODUCTION

In the chemical and mineral industries, particle-size reduction is often one of the most costly of the operations involved; and, frequently, an appreciable part of the cost is the result of inefficient operation or the use of the wrong type of equipment. This is due, largely, to: (1) the very great number of different crushers and grinders on the market, (2) the great differences in the properties of solids which are subjected to size-reduction, (3) the wide differences in the requirements for the finished products, and (4) the highly empirical nature of our whole knowledge of crushing and grinding.

Theories of Particle-size Reduction. Particle-size reduction is an energy consuming operation, and the amount of energy required increases greatly as the particle size of the product is decreased. Unfortunately, attempts to express the relationship between energy consumption and particle-size reduction mathematically led to the development of two relationships, known, respectively, as *Rittinger's* and *Kick's laws*, and this precipitated a time-consuming controversy that has raged for many years.

Rittinger's Law. This law states that the energy required for crushing is proportional to the new surface formed. However, since the total surface in a given weight of material is inversely proportional to the particle size, Rittinger's law is usually expressed as

$$E = c \left(\frac{1}{D_2} - \frac{1}{D_1} \right) \quad (8.1)$$

where

E = energy required for crushing.

c = proportionality constant.

D_1 = particle size before crushing.

D_2 = particle size after crushing.

Kick's Law. According to this law, the energy required to crush a given quantity of material to a desired fraction of the original particle size is constant regardless of the original size of the feed. This is expressed mathematically as

$$E = k \log \frac{D_1}{D_2} \quad (8.2)$$

Although it is true that in ordinary work neither Rittenger's nor Kick's law expresses the facts, careful experiments have demonstrated, beyond doubt, the theoretical validity of Rittenger's law, and have clearly shown that Kick's law has neither practical nor theoretical significance.⁴

Energy Consumption for Size Reduction. In Equations (8.1) and (8.2), the energy E is the difference between the energy consumptions of the machine while operating under load and while running idle. This does not imply, however, that all of the net energy is utilized in doing useful work; on the contrary, most of it is wasted in overcoming friction between the particles of solid and in causing deformation without breaking.¹³ When this wasted energy is added to the energy used in overcoming friction of the machine itself, it is found that the actual energy consumed in creating new surface is an extremely small fraction of the total energy input to the machine, amounting to approximately 1.3 per cent in roll crushing³ and $\frac{1}{10}$ th of one per cent in ball milling,¹² while even in drop-weight machines the efficiency is only of the order of 1 to 3 per cent.⁵

Although these figures are somewhat misleading because they do not take into consideration the fact that the force must act through a distance before new surface is produced, they demonstrate very clearly that size reduction is extremely inefficient.

From a practical point of view, the theoretical efficiency, as such, is of little importance, for it is the cost per unit of product that is ordinarily the controlling factor. To keep the cost to a minimum, consideration must be given to: (1) the selection of the proper type of equipment for the given job; (2) the use of closed-circuit grinding to remove the fines as fast as they are formed and thus prevent the consumption of large amounts of energy in grinding the product to a smaller size than necessary; (3) the use of wet grinding, where practical; (4) drying the feed if it contains excessive amounts of moisture and wet grinding is impractical; and (5) the use of dry-grinding aids such as have proved effective in reducing the cost of grinding cement clinker.¹¹

Crushing and Grinding Equipment. Equipment for size reduction may be classified into primary crushers, secondary crushers, and fine grinders; some of the most important mills of each type are:^{1, 2, 6, 14, 15}

Primary Crushers

Jaw crushers.

Gyratory crushers.

Single-roll crushers.

Secondary Crushers

Rolls.

Disk crushers.

Hammer mills.

Stamp mills.

Fine Grinders

Ball, tube, and rod mills.

Roller mills.

Ring roll pulverizers.

Bowl mills.

Ball-bearing type pulverizers.

The line of demarkation between primary crushers, secondary crushers, and fine grinders is by no means sharp, for some equipment falls into more than one classification. For example, although the disk type of machine is usually considered to be a secondary crusher, some disk machines are used for coarse crushing while others are used for pulverizing.

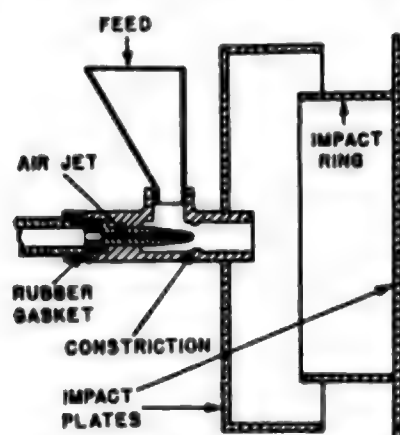


Figure 1. Jet-Type Pulverizer

Within recent years, interest has been revived in the jet mill, a pulverizer which is radically different from any of those listed above. In this machine, which is shown diagrammatically in Figure 1, the feed is ejected from a nozzle at high velocity by means of steam or air and is shattered by impact against a target.^{7, 16}

Mills of this type are now on the market; they are capable of pulverizing materials to a very small size, but they are expensive to operate. However, it is probable that as they are improved in efficiency their use will become much more wide-spread.

Grindability of Materials. In order to estimate the energy required for a given size reduction, it is necessary to know the absolute or the relative *grindability* of the material. Grindability is defined in terms of either: (1) the Rittinger constant expressed as square centimeters of surface produced per foot-pound of energy expended,⁸ or (2) the tons per hour that can be pulverized to a definite fineness in a given pulverizer.⁹ Relative grindability, obviously, is the ratio of the grindability of a substance to the grindability of a standard substance. Considerable progress has been made in determining the grindability of materials, and a list of the grindabilities of a number of materials has been prepared by Hardgrove.¹⁰ However, in using such data it must be realized that the grindability of a substance depends upon the type of pulverizer used and that a particular pulverizer may operate much more efficiently at

one grindability than at another. Nevertheless, as more grindability data become available, the selection of crushing and grinding equipment will be placed on a much more rational basis than at present.

Laboratory Crushing and Grinding Work. Crushing and grinding experiments have a definite place in both instructional and industrial chemical engineering laboratories. For instructional purposes, experiments involving measurement of the energy required for various size reductions and for various materials emphasize the great increase in energy requirement as the particle size decreases, and illustrate the vast differences in the grindability of different materials. Such experiments also impart valuable experience in the determination of size distribution and in the calculation of surface area.

In the industrial laboratory, a few crushing and grinding tests may point the way to lower production costs by indicating the most efficient method for carrying out a given grinding job.

NOMENCLATURE

- c = proportionality constant in Rittinger's law.
- D_1 = particle size before crushing.
- D_2 = particle size after crushing.
- E = energy required for crushing.
- k = proportionality constant in Kick's law.

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A JAW CRUSHER MODIFIED FOR EXPERIMENTAL PURPOSES

By

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Description. In order to permit the measurement of power consumption during crushing, a small 3-inch jaw crusher was fitted with a $\frac{1}{2}$ -H.P. electric motor, which in turn was connected to a 110 volt A.C. line through a wattmeter.

As shown in Figure 1, the crusher is mounted on a supporting framework made of steel angles, and the motor is suspended beneath it on a platform hinged in such a manner that the weight of the motor plus the platform maintains a uniform tension on the drive belt, which operates in a V-groove cut in one of the fly wheels. This arrangement also acts as a safety device to protect the crusher and motor in case tramp iron or any other non-crushable material enters with the feed. To facilitate attachment of the wattmeter, a connection panel is mounted directly to the crusher assembly.

Materials and Cost. Except for the cost of the motor, which was \$14, the cost of the materials for this modification was almost negligible—approximately \$3.75. The complete bill of materials exclusive of the crusher and the wattmeter is given in Table 1. The crusher was an old piece of equipment that was available, and the wattmeter is used for a variety of experiments and is not a permanent part of this setup.

Notes on Construction. The method of mounting the crusher and the motor is evident from Figure 1. The steel angles and the strap steel for the framework were cut to dimensions with an acetylene torch, the appropriate holes for the bolts and motor-pivot rod were drilled, and the framework was welded together. The motor platform was made in a similar manner. The groove in the fly wheel was cut on a lathe.

No dimensions are given here for, obviously, they will depend upon the size, type, and make of crusher available.

Operation. In performing a crushing experiment, the power input is first measured while the crusher is running idle and then while the feed is introduced at a uniform rate. As the power input fluctuates over rather wide limits while the crusher is operating under load, it is necessary to take frequent readings of the wattmeter and to plot them against time in order to determine the mean value.

TYPICAL EXPERIMENT

Objects. 1. To determine the energy consumption as a function of size reduction for the crushing of: (a) marble, and (b) beryl.

2. To compare the experimentally determined values with those predicted by Kick's and Rittinger's laws.

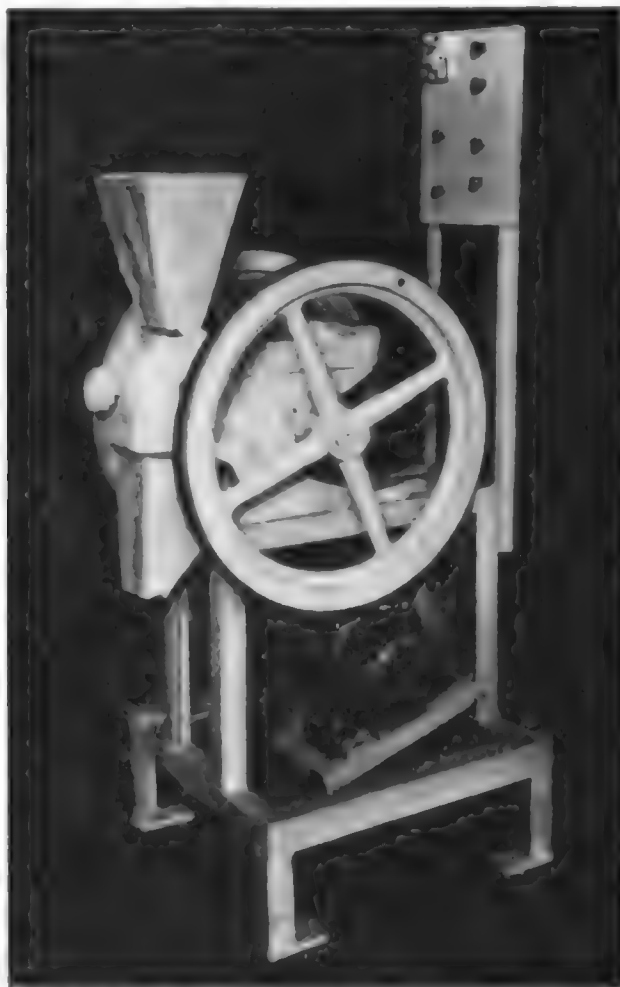


Figure 1. Jaw Crusher Modified for Experimental Purposes—University of New Hampshire

Procedure. 1. Select several pounds of marble having individual particle sizes of about one inch, and screen it through a set of sieves to determine the size distribution.

2. Adjust the jaw of the crusher so as to have a rather wide opening, start the motor, and read the wattmeter.

3. Introduce a weighed quantity of feed at a uniform rate sufficient to keep the jaws filled. During the run, read the wattmeter as rapidly as possible at timed intervals. Readings should be made at least every two or three seconds because of the fluctuation in the wattage.

4. Collect the product for the entire run, and sample it by quartering to obtain about a pound for size determination.

5. Screen the sample through a set of standard sieves, determine the size distribution, and plot the cumulative screen analysis.

6. Repeat the experiment using the same feed but with the jaws set closer together.

TABLE 1
Bill of Materials for Modification of Jaw Crusher
University of New Hampshire

Quantity	Description	Size	Material
2 ft. 8 in.	Angles	$1\frac{1}{2} \times 1\frac{1}{2} \times \frac{1}{8}$ in.	Steel
10 ft.	"	$1\frac{1}{2} \times 1\frac{1}{2} \times \frac{1}{8}$ in.	"
2 ft. 2 in.	"	$1 \times 1 \times \frac{1}{2}$ in.	"
$4\frac{1}{2}$ ft.	"	$\frac{3}{4} \times \frac{3}{4} \times \frac{1}{2}$ in.	"
$2\frac{1}{2}$ ft.	Flats	$1\frac{1}{2} \times \frac{1}{8}$ in.	"
$1\frac{1}{2}$ sq. ft.	Sheet	24 gage	"
1	Panel	$8 \times 10 \times \frac{1}{8}$ in. thick	Transite
1	Rod	$\frac{1}{2}$ in. diam. \times $7\frac{1}{2}$ in. long	Steel
6	Binding post	—	Brass
1	Single-pole, single-throw switch	—	Copper
6	Stove bolts	$\frac{1}{8}$ in. \times $\frac{1}{2}$ in. long	Steel
3	Bolts and nuts	$\frac{1}{2}$ in. — 13 NC \times $1\frac{1}{2}$ in. long	"
4	" " "	$\frac{1}{2}$ in. — 20 NC \times 1 in. long	"
1	Electric motor	$\frac{1}{2}$ H.P., 60 cycle, 110 volts, 1725 R.P.M.	—
1	V-belt pulley	3 in. diam., for $\frac{1}{2}$ in. diam. shaft	Cast iron
1	V-belt	$\frac{1}{2}$ in. wide \times 56 in. circumference	Rubberized cotton
40 ft.	Electric wire	No. 14 gage	Copper

7. Repeat the experiment using beryl as the feed and with approximately the same two jaw openings.

8. For each run, calculate the average particle size and surface area before and after crushing, and calculate the theoretical power consumption according to Kick's and Rittinger's laws.

9. Compare the experimentally determined values with those calculated.

EXPERIMENTAL RESULTS AND CALCULATIONS

A summary of the experimental and calculated results for the crushing of marble and beryl is given in Table 2. These results were obtained in the following manner:

Preparation of the Feed. Some large pieces of marble and beryl were crushed in a large jaw crusher and screened on 1-inch, $\frac{3}{4}$ -inch, and $\frac{1}{2}$ -inch screens. Sufficient marble with a size of $\frac{3}{4}$ to 1 inch was obtained for

TABLE 2
Summary of Results for the Crushing of Marble and Beryl

Item No.	Item	Run number			
		1	2	3	4
	Feed:				
	Material	Marble	Marble	Beryl	Beryl
1	Quantity, grams	3500	2997	2970	2974
	Size distribution, %				
	through retained on				
2	1-in. $\frac{3}{4}$ -in.	100	100	87.6	87.6
3	$\frac{3}{4}$ -in. $\frac{1}{2}$ -in.	0	0	32.4	32.4
4	Average screen size, in.....	0.875	0.875	0.8025	0.8025
5	Total fictitious surface per 100 lb.	114.4	114.4	124.3	124.3
	Product:				
6	Average particle size, in.....	0.01257	0.00783	0.01229	0.01892
7	Total fictitious surface per 100 lb.	7,989	12,793	8,138	5,291
8	Throat opening of jaws, in.....	0.25	0.125	0.125	0.25
9	Time of run, sec.....	215	303	519	222
	Power consumption, watts:				
10	(a) without load.....	205	205	205	205
11	(b) with load (mean).....	235	238	340	366
12	(c) net for crushing.....	30	33	135	161
	Energy consumption, kw.-hr./ton:				
13	(a) total.....	3.65	5.06	14.73	6.86
14	(b) net.....	0.465	0.843	5.845	3.036
15	Cost of crushing @ \$0.01 per kw.-hr., \$/ton..	0.0365	0.0806	0.1473	0.0686

the experiment, but it was necessary to blend some of the $\frac{1}{2}$ - to $\frac{3}{4}$ -inch beryl fraction with the $\frac{3}{4}$ - to 1-inch fraction to obtain sufficient feed (Items 2 and 3, Table 2). Each quantity of feed was divided into two portions so that two runs could be made with each.

Determination of the Power Input. For each run, the jaws of the crusher were set for the desired throat opening (Item 8), the motor was turned on, the wattmeter was read with the crusher running idle (Item 10), and then feed was introduced into the crusher. A stop watch was started when feeding was begun and stopped when the last of the

product left the crusher (Item 9). During this period, the wattmeter was read as rapidly as possible and these readings were later plotted as a function of time in Figure 2. The curves so obtained were graphically integrated and the values of the integrals were divided by the total time to obtain the mean power consumption with load (Item 11).

For example, for Run No. 1 the area under the curve was 50,500 watt-seconds and the time of the run was 215 seconds; therefore, the mean power consumption with load was

$$\frac{50,500}{215} = 235 \text{ watts}$$

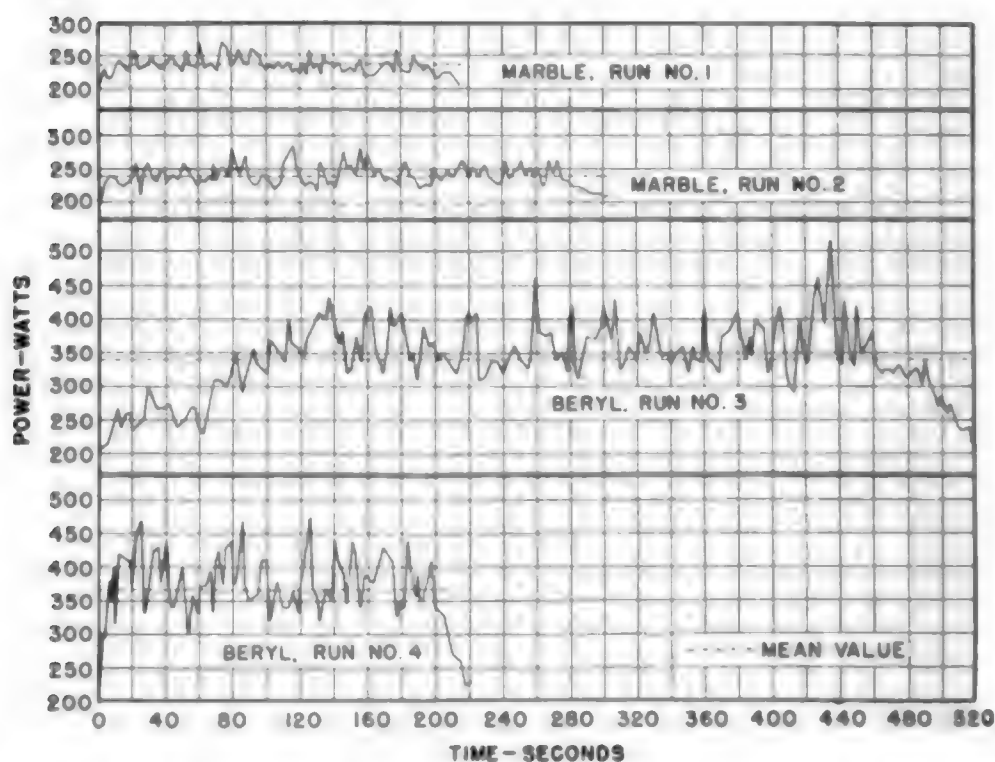


Figure 2. Log of the Power Consumption During the Crushing of Marble and Beryl

and the net energy consumption (Item 14) was

$$\frac{(235 - 205)}{1000} \times \frac{215}{3600} \times \frac{454}{3500} \times 2000 = 0.465 \text{ kw.-hr./ton}$$

where

$(235 - 205) = 30 = \text{net energy consumption, watts (Item 12).}$

$1000 = \text{watts/kw.}$

$215 = \text{time of run, sec. (Item 9).}$

$3600 = \text{sec./hr.}$

454 = grams/lb.

3500 = grams of product (Item 1).

2000 = lb./ ton.

Calculation of the Surface Area and the Mean Particle Size. To obtain the total surface area of the product from each run and the mean

TABLE 3
Screen Analysis of Products

Material.....		Marble				Beryl			
Run No.....		1		2		3		4	
Screen No., U. S. Standard Series	Screen open- ing, in.	Per cent retained on screen				Per cent retained on screen			
		Differ- ential	Cumu- lative	Differ- ential	Cumu- lative	Differ- ential	Cumu- lative	Differ- ential	Cumu- lative
—	0.500	0	0	0	0	0	0	0	0
—	0.375	0.84	0.84	0	0	0	0	0.84	0.84
—	0.313	5.58	6.42	0	0	0	0	7.38	8.22
—	0.250	16.65	23.07	0	0	1.18	1.18	18.00	26.22
4	0.187	6.03	29.10	1.38	1.38	3.51	4.69	8.38	34.60
6	0.132	6.98	36.08	7.33	8.71	18.26	22.95	12.10	46.70
8	0.0937	3.74	39.82	7.17	15.88	12.35	35.30	9.36	56.06
10	0.0787	1.31	41.13	1.65	17.53	6.58	41.86	3.88	59.94
12	0.0661	1.11	42.24	1.55	19.03	3.81	45.69	3.93	63.87
16	0.0469	2.63	44.87	2.69	21.77	9.92	55.61	7.68	71.55
18	0.0394	1.34	46.21	0.84	22.61	4.17	59.78	2.47	74.02
20	0.0331	1.35	47.56	1.24	23.85	4.28	64.06	3.35	77.37
30	0.0232	3.03	50.59	3.13	26.98	6.35	70.41	4.50	81.87
40	0.0165	5.89	56.48	6.24	33.22	6.52	76.93	4.21	86.08
50	0.0117	12.85	69.33	16.77	49.99	4.45	81.38	2.98	89.06
60	0.0098	4.58	73.91	13.75	63.74	2.10	83.48	1.29	90.35
80	0.0070	13.56	87.47	15.14	78.88	3.78	87.26	2.32	92.67
100	0.0059	4.09	91.56	5.86	84.74	2.65	89.91	1.56	94.23
120	0.0049	1.47	93.03	3.84	88.58	1.10	91.01	0.65	94.88
140	0.0041	1.34	94.37	1.31	89.89	0.91	91.92	0.67	95.55
200	0.0029	2.59	96.56	3.95	93.84	2.90	94.82	1.32	96.87
325	0.0017	1.62	98.58	3.04	96.88	2.29	97.11	1.22	98.09
Through 325		1.42	100.00	3.12	100.00	2.89	100.00	1.91	100.00

particle size, the product was first weighed (Item 1), and then screened through a set of standard sieves; the results are given in Table 3 and Figure 3. Calculations were then made as follows:

If each particle were a perfect cube with a linear dimension D equal to that of the screen opening, its area A_p would be:

$$A_p = 6D^2 \quad (1)$$

and its volume V_p would be:

$$V_p = D^3 \quad (2)$$

Although the particles are not cubes, it can be assumed that on the average their areas and volumes are proportional to the areas and vol-

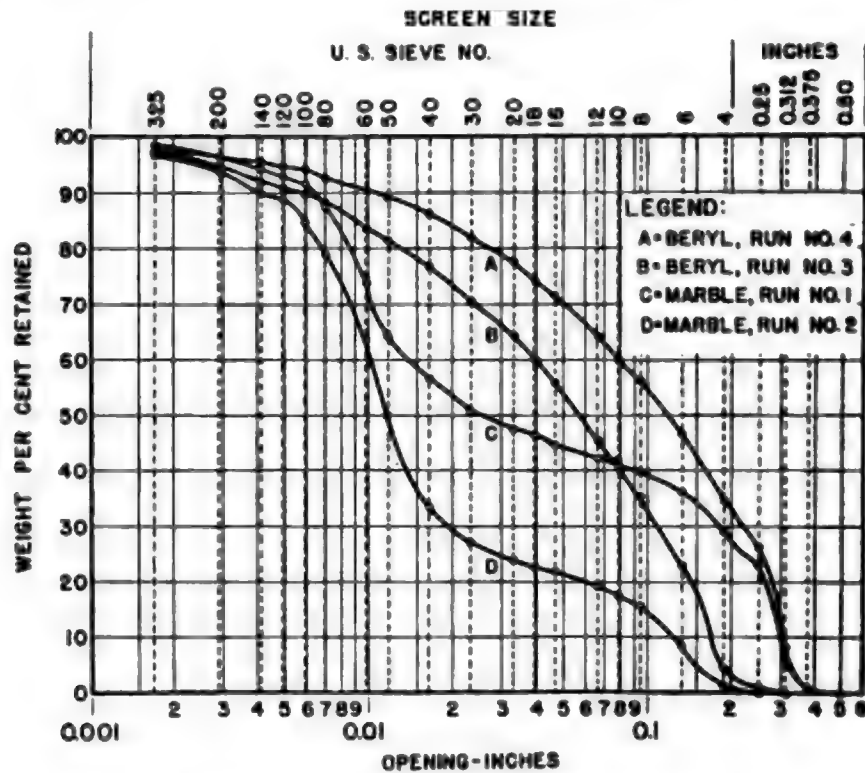


Figure 3. Screen Analyses of Crushed Marble and Beryl

umes of cubes of the same screen size; or

$$A_p = aD^2 \quad (3)$$

and

$$V_p = bD^3 \quad (4)$$

where a and b are constants for a given material.

The number of particles N in a uniform-sized fraction of weight w is

$$\begin{aligned} N &= \frac{w}{\rho} \times \frac{1}{V_p} \\ &= \frac{w}{\rho} \times \frac{1}{bD^3} \\ &= \frac{w}{cD^3} \end{aligned} \quad (5)$$

where ρ is the density, and c equals $b\rho$.

and the total area A_f of all the particles in the fraction is

$$A_f = NA_p \quad (6)$$

$$\begin{aligned} &= \frac{w}{cD^3} (aD^2) \\ &= \frac{a}{c} \times \frac{w}{D} \end{aligned} \quad (7)$$

This equation may also be written as

$$A'_f = \frac{w}{D} \quad (8)$$

where A'_f is a fictitious surface defined by the equation

$$A'_f = A_f \frac{c}{a} \quad (9)$$

The total fictitious surface area A' of all the particles in the product is, therefore,

$$A' = \Sigma A'_f = \Sigma \frac{w}{D} \quad (10)$$

and the mean particle size D_m in a total weight of product W is

$$D_m = \frac{\Sigma w}{A'} = \frac{W}{A'} \quad (11)$$

The calculations involved in determining the area and mean particle size of the products (Items 6 and 7) are summarized in Table 4. The calculations for the feed (Items 4 and 5) were made in a similar manner.

Calculation of the Relative Energy Consumption According to Rittinger's and Kick's Laws.^{1,2} 1. *Rittinger's Law.* According to this law, the energy consumed in crushing is proportional to the new surface formed; or

$$E = C (A_2 - A_1) \quad (12)$$

where

E = energy consumed in crushing.

A_2 = total surface area of the product.

A_1 = total surface area of the feed.

C = proportionality constant.

As the area is proportional to the reciprocal of the particle size, this equation can also be written as

$$E = C' \left(\frac{1}{D_{m_2}} - \frac{1}{D_{m_1}} \right) \quad (13)$$

where D_m and D_{m_1} are the mean particle sizes of the product and feed, respectively.

TABLE 4
Summary of Calculations of Surface Area and Mean Particle Size

Screen Analysis		D, average screen size,* in.	Marble				Beryl			
Through	Retained on		Run No. 1		Run No. 2		Run No. 3		Run No. 4	
			w, per cent = lb./ 100 lb.	$A'_f = \frac{w}{D}$	w, per cent = lb./ 100 lb.	$A'_f = \frac{w}{D}$	w, per cent = lb./ 100 lb.	$A'_f = \frac{w}{D}$	w, per cent = lb./ 100 lb.	$A'_f = \frac{w}{D}$
3/4-inch	3/4-inch	0.4375	0.84	1.9	0	0	0	0	0.84	1.9
3/4-inch	3/4-inch	0.3438	5.58	16.3	0	0	0	0	7.38	21.4
3/4-inch	3/4-inch	0.2812	16.66	59.3	0	0	1.18	4.2	18.00	64.1
3/4-inch	4 mesh	0.2185	6.03	27.6	1.38	6.3	3.51	16.6	8.38	38.3
4 mesh	6 mesh	0.1595	6.98	43.7	7.33	45.9	18.26	114.6	12.10	75.8
6 mesh	8 mesh	0.1128	3.74	33.2	7.17	63.5	12.35	109.8	9.36	83.1
8 mesh	10 mesh	0.0862	1.31	15.2	1.65	19.2	6.58	76.4	3.88	45.0
10 mesh	12 mesh	0.0724	1.11	15.3	1.55	21.4	3.81	52.6	3.93	54.4
12 mesh	16 mesh	0.0565	2.63	46.5	2.69	47.6	9.92	175.5	7.68	135.9
16 mesh	18 mesh	0.0432	1.34	31.0	0.84	19.5	4.17	96.5	2.47	57.3
18 mesh	30 mesh	0.0363	1.36	37.2	1.24	34.1	4.28	118.0	3.35	94.9
30 mesh	30 mesh	0.0281	3.03	107.8	3.13	111.4	6.35	226.0	4.50	160.0
30 mesh	40 mesh	0.0199	5.89	296.0	6.24	313.2	6.53	327.5	4.21	211.9
40 mesh	50 mesh	0.0141	12.85	912.0	16.77	1,190.0	4.45	315.8	2.98	211.0
50 mesh	60 mesh	0.0107	4.58	428.5	13.75	1,285.0	2.10	196.1	1.29	130.5
60 mesh	80 mesh	0.0084	13.56	1,615	15.14	1,802.0	3.78	449.5	2.32	376.0
80 mesh	100 mesh	0.0064	4.09	539.2	5.86	916.0	2.65	414.2	1.56	344.0
100 mesh	120 mesh	0.0054	1.47	270.2	3.84	711.0	1.10	207.2	0.65	130.5
120 mesh	140 mesh	0.0045	1.34	297.6	1.31	290.9	0.91	202.0	0.67	149.0
140 mesh	200 mesh	0.0035	2.59	740.0	3.95	1,129.0	2.90	828.5	1.32	377.0
200 mesh	325 mesh	0.0023	1.62	704.0	3.04	1,322.0	2.29	995.3	1.22	531.0
325 mesh		0.0009	1.42	1,578.0	3.12	3,465.0	2.89	3,212.0	1.91	2,118.0
$A' = \text{total surface}$				7,969.4		12,793.0		8,138.0		5,290.9
$D_m = \text{mean particle size, inches} = \frac{100}{A'}$				0.01257		0.00783		0.01226		0.01092

* Average opening of screen through and screen retained on.

The ratio between the energy consumption for the same material under two different conditions of size reduction, therefore, is

$$\frac{E_g}{E_l} = \frac{(A'_2 - A'_1)_g}{(A'_2 - A'_1)_l} = \frac{\left(\frac{1}{D_{m_2}} - \frac{1}{D_{m_1}}\right)_g}{\left(\frac{1}{D_{m_2}} - \frac{1}{D_{m_1}}\right)_l} \quad (14)$$

where the subscripts g and l refer to the conditions of greater and less reduction, respectively.

Substituting the values of A' and D_m obtained experimentally, (Tables 4 and 2) the calculations are as follows

$$\begin{aligned} a. \text{ For marble, } \frac{E_o}{E_i} &= \frac{12,793 - 114.4}{7,969 - 114.4} = \frac{\frac{1}{0.00783} - \frac{1}{0.875}}{\frac{1}{0.01257} - \frac{1}{0.875}} \\ &= 1.614 \end{aligned}$$

$$\begin{aligned} b. \text{ For beryl, } \frac{E_o}{E_i} &= \frac{8,138 - 124.3}{5,291 - 124.3} = \frac{\frac{1}{0.01229} - \frac{1}{0.8025}}{\frac{1}{0.01892} - \frac{0}{0.8025}} \\ &= 1.550 \end{aligned}$$

2. *Kick's Law.* According to this law, the energy consumed in crushing is proportional to the decrease in volume; or, expressed mathematically,

$$dE = -k \frac{dV}{V} \quad (15)$$

Therefore, the total energy consumed E for a reduction in volume of V_1 to V_2 is

$$E = \int_{V_1}^{V_2} dE = -k \int_{V_1}^{V_2} \frac{dV}{V} \quad (16)$$

$$= k' \log \frac{V_1}{V_2} \quad (17)$$

Since volume is proportional to the cube of the diameter, Equation (17) may be written as

$$E = k' \log \frac{D_1^3}{D_2^3} \quad (18)$$

$$= 3k' \log \frac{D_1}{D_2} \quad (19)$$

$$= k'' \log \frac{D_1}{D_2} \quad (20)$$

which is the usual expression of Kick's law.

According to Equation (20), it requires the same energy consumption for a size reduction of 2 inch to 1 inch as it does for a reduction of 1 inch to $\frac{1}{2}$ inch, or $\frac{1}{2}$ inch to $\frac{1}{4}$ inch, etc. The ratio between the energy consumption for two different size reductions, therefore, is

$$\frac{E_s}{E_t} = \frac{\log \left(\frac{D_1}{D_2} \right)_s}{\log \left(\frac{D_1}{D_2} \right)_t} \quad (21)$$

Substituting the experimental values in Equation (21) gives:

$$\begin{aligned} a. \text{ For marble, } \frac{E_s}{E_t} &= \frac{\log \frac{0.875}{0.00783}}{\log \frac{0.875}{0.01257}} \\ &= 1.070 \end{aligned}$$

TABLE 5

Comparison of Relative Energy Consumption, $\frac{E_s}{E_t}$

	Experimental	Rittenger's law	Kick's law
Marble.....	1.810	1.614	1.070
Beryl.....	1.926	1.550	1.065

$$\begin{aligned} b. \text{ For beryl, } \frac{E_s}{E_t} &= \frac{\log \frac{0.8025}{0.01229}}{\log \frac{0.8025}{0.01892}} \\ &= 1.065 \end{aligned}$$

Comparison of the Experimental and the Calculated Relative Energy Consumptions. The ratio between the experimental net energy consumptions for each set of runs is:

$$\begin{aligned} 1. \text{ For marble, } \frac{E_s}{E_t} &= \frac{(\text{kw.-hr./ton})_s}{(\text{kw.-hr./ton})_t} = \frac{0.842}{0.465} = 1.810 \\ 2. \text{ For beryl, } \frac{E_s}{E_t} &= \frac{(\text{kw.-hr./ton})_s}{(\text{kw.-hr./ton})_t} = \frac{5.845}{3.036} = 1.926 \end{aligned}$$

Table 5 shows a comparison of these values with those calculated by means of Rittenger's and Kick's laws.

It is evident from Table 5 that, although the experimental results are appreciably different from those calculated, Rittenger's law con-

forms much more closely to the actual conditions of this experiment than does Kick's law.

NOMENCLATURE

- A = surface area.
 A' = fictitious surface area.
 D = diameter.
 E = energy consumed in crushing.
 N = number of particles.
 V = volume.
 W = total weight of product.
 w = weight of uniform-sized fraction
 ρ = density.
 $a, b, c, C, C', k, k' \text{ and } k''$ = constants

Subscripts

- f = fraction.
 g = greater reduction.
 l = lesser reduction.
 m = mean value.
 p = particle.
 1 = feed.
 2 = product.

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TABLE 1
Conversion Factors

Multiply	By	To Obtain
Atmospheres.....	76.0	Cm. of Hg
“.....	29.92	In. of Hg
“.....	33.90	Ft. of water
“.....	14.70	lb./sq. in.
B.t.u.....	0.2520	Kg.-cal.
“.....	777.5	Ft.-lb.
“.....	3.927×10^{-4}	H.P.-hr.
“.....	2.928×10^{-4}	Kw.-hr.
B.t.u.		Gram cal.
hr. \times ft. ³ \times °F.....	0.0001355	sec. \times cm. ³ \times °C.
B.t.u.-ft.		Gram cal.-cm.
hr. \times ft. ³ \times °F.....	0.00413	sec. \times cm. ³ \times °C.
B.t.u./min.....	12.96	Ft.-lb./sec.
“ / “.....	0.02356	Horsepower
“ / “.....	0.01757	Kilowatts
Calories (gram).....	3.968×10^{-4}	B.t.u.
“ “.....	3.087	Ft.-lb.
“ “.....	4.130×10^{-4}	Liter-atm.
“ “.....	4.185	Joules
Cal. (gram)		B.t.u.
sec. \times cm. ³ \times °C.....	7,380	hr. \times ft. ³ \times °F.
Cal. (gram)-cm.		B.t.u.-ft.
sec. \times cm. ³ \times °C.....	242.1	hr. \times ft. ³ \times °F.
Centimeters.....	0.3937	Inches
Cm. of Hg.....	0.01316	Atmospheres
“ “ “.....	0.4461	Ft. of water
“ “ “.....	27.85	Lb./sq. ft.
“ “ “.....	0.1934	Lb./sq. in.
Cm./sec.....	1.969	Ft./min.
“ / “.....	0.03281	Ft./sec.
Cm./sec./sec.....	0.03281	Ft./sec./sec.
Centipoises.....	0.000672	Lb./(ft. \times sec.)
“.....	2.42	Lb./(ft. \times hr.)
Cubic cm.....	3.531×10^{-6}	Cubic ft.
“ “.....	6.102×10^{-3}	Cubic in.
“ “.....	2.642×10^{-4}	Gallon
Cubic ft.....	2.832×10^4	Cubic cm.
“ “.....	1,728	Cubic in.
“ “.....	7.48052	Gallon
“ “.....	28.32	Liters
Cubic ft./min.....	472.0	Cubic cm./sec.
“ “ / “.....	0.1247	Gal./sec.
“ “ / “.....	0.4720	Liters/sec.
Cubic ft./sec.....	448.831	Gal./min.
Cubic in.....	16.39	Cubic cm.
“ “.....	5.787×10^{-4}	Cubic ft.

TABLE 1—Continued

Multiply	By	To Obtain
Feet.....	30.48	Centimeters
Ft. of water.....	0.02950	Atmospheres
" " ".....	0.8826	In. of Hg
" " ".....	62.43	Lb./sq. ft.
" " ".....	0.4335	Lb./sq. in.
Ft./min.....	0.5080	Cm./sec.
" / ".....	0.01667	Ft./sec.
Ft./sec./sec.....	30.48	Cm./sec./sec.
Ft.-lb.....	1.286×10^{-3}	B.t.u.
" - ".....	5.050×10^{-7}	H.P.-hr.
" - ".....	3.241×10^{-4}	Kg.-cal.
" - ".....	3.766×10^{-7}	Kw.-hr.
Ft.-lb./min.....	3.030×10^{-6}	Horsepower
" - " / ".....	2.260×10^{-6}	Kilowatts
Gallons.....	3,785	Cubic cm.
".....	0.1337	Cubic ft.
".....	231	Cubic in.
Gallons of water.....	8.3453	Lb. of water
Gallons/min.....	2.228×10^{-3}	Cubic ft./sec.
Grams.....	980.7	Dynes
".....	15.43	Grains
".....	0.03527	Oz. (avoir.)
".....	0.03215	Oz. (troy)
".....	2.205×10^{-3}	Pounds
Grams/liter.....	58.417	Grains/gal.
" / ".....	8.345	Lb./1000 gal.
" / ".....	0.062427	Lb./cu. ft.
Horse-power.....	42.44	B.t.u./min.
" ".....	33,000	Ft.-lb./min.
" ".....	550	Ft.-lb./sec.
" ".....	10.70	Kg.-cal./min.
" ".....	0.7457	Kilowatts
H.P. (boiler).....	33,479	B.t.u./hr.
" ".....	9.803	Kilowatts
H.P.-hr.....	2,547	B.t.u.
" ".....	1.98×10^4	Ft.-lb.
" ".....	641.7	Kg.-cal.
" ".....	0.7457	Kw.-hr.
Inches.....	2.540	Centimeters
In. of Hg.....	0.03342	Atmospheres
" " ".....	1.133	Ft. of water
" " ".....	70.73	Lb./sq. ft.
" " ".....	0.4912	Lb./sq. in.
In. of water.....	0.002458	Atmospheres
" " ".....	0.07355	In. of Hg
" " ".....	5.202	Lb./sq. ft.
" " ".....	0.03613	Lb./sq. in.

TABLE 1—Continued

Multiply	By	To Obtain
Kilograms.....	980.665×10^3	Dynes
".....	2.205	Pounds
Kg.-cal.....	3.968	B.t.u.
" - ".....	0.0011628	Kw.-hr.
" - ".....	3,087.4	Ft.-lb.
Kg.-cal./min.....	0.09351	Horsepower
" " / ".....	0.00972	Kilowatts
Kilowatts.....	56.92	B.t.u./min.
".....	4.425×10^4	Ft.-lb./min.
".....	1.341	Horsepower
".....	14.34	Kg.-cal./min.
Kw.-hr.....	3,415	B.t.u.
" ".....	2.655×10^6	Ft.-lb.
" ".....	1.341	H.P.-hr.
" ".....	860.5	Kg. cal.
Liters.....	0.03531	Cubic ft.
".....	61.02	Cubic in.
".....	0.2642	Gallons
Liters/min.....	5.886×10^{-4}	Cubic ft./sec.
" / ".....	4.403×10^{-3}	Gal./sec.
Miles.....	1.609×10^3	Centimeters
".....	5,280	Feet
".....	1,760	Yards
Miles/hr.....	44.70	Cm./sec.
" / ".....	1.467	Ft./sec.
" / ".....	0.8684	Knots
Millimeters.....	0.03937	Inches
Mg./liter.....	1	Parts/million
Parts/million.....	0.0584	Grains/U.S. Gal.
" / ".....	8.345	Lb./million gal.
Pounds.....	16	Oz. (avoir.)
".....	14.5833	Oz. (troy)
".....	7,000	Grains
".....	453.5924	Grams
Pounds of water.....	0.01602	Cubic ft.
" " ".....	27.68	Cubic in.
Pounds water/min.....	2.670×10^{-4}	Cubic ft./sec.
Pounds/cu. ft.....	0.016018	Grams/cu. cm.
Pounds/sq. ft.....	0.01602	Ft. of water
" / " ".....	6.945×10^{-3}	Lb./sq. in.
Pounds/sq. in.....	0.06804	Atmospheres
" / " ".....	2.307	Ft. of water
" / " ".....	2.036	In. of Hg
Square inches.....	6.452	Sq. cm.
" ".....	6.944×10^{-3}	Sq. ft.
Square miles.....	640	Acres

TABLE 1—*Concluded*

Multiply	By	To Obtain
Tons (long).....	2,240	Pounds
" (long).....	1.12000	Tons (short)
Tons (short).....	2,000	Pounds
" (short).....	0.89287	Tons (long)
Watts.....	0.05692	B.t.u./min.
".....	44.26	Ft.-lb./min.
".....	1.341×10^{-3}	Horsepower
".....	0.01434	Kg.-cal./min.
".....	10^{-3}	Kilowatts
Watt-hours.....	3.415	B.t.u.
" ".....	2,655	Ft.-lb.
" ".....	1.341×10^{-3}	H.P.-hr.
" ".....	0.8605	Kg.-cal.
" ".....	10^{-3}	Kw.-hr.

TABLE 2
Temperature Conversions

°C.	Temp. in °C. or °F. to be converted	°F.	°C.	Temp. in °C. or °F. to be converted	°F.	°C.	Temp. in °C. or °F. to be converted	°F.
-17.8	0	32	1.7	35	95.0	26.7	80	176.0
-17.2	1	33.8	2.2	36	96.8	27.8	82	179.6
-16.7	2	35.6	2.8	37	98.6	28.9	84	183.2
-16.1	3	37.4	3.3	38	100.4	30.0	86	186.8
-15.6	4	39.2	3.9	39	102.2	31.1	88	190.4
-15.0	5	41.0	4.4	40	104.0	32.2	90	194.0
-14.4	6	42.8	5.0	41	105.8	33.3	92	197.6
-13.9	7	44.6	5.6	42	107.6	34.4	94	201.2
-13.3	8	46.4	6.1	43	109.4	35.6	96	204.8
-12.8	9	48.2	6.7	44	111.2	36.7	98	208.4
-12.2	10	50.0	7.2	45	113.0	37.8	100	212.0
-11.7	11	51.8	7.8	46	114.8	38.9	102	215.6
-11.1	12	53.6	8.3	47	116.6	40.0	104	219.2
-10.6	13	55.4	8.9	48	118.4	41.1	106	222.8
-10.0	14	57.2	9.4	49	120.2	42.2	108	226.4
-9.4	15	59.0	10.0	50	122.0	43.3	110	230.0
-8.9	16	60.8	10.6	51	123.8	44.4	112	233.6
-8.3	17	62.6	11.1	52	125.6	45.6	114	237.2
-7.8	18	64.4	11.7	53	127.4	46.7	116	240.8
-7.2	19	66.2	12.2	54	129.6	47.8	118	244.4
-6.7	20	68.0	12.8	55	131.0	48.9	120	248.0
-6.1	21	69.8	13.3	56	132.8	50.0	122	251.6
-5.6	22	71.6	13.9	57	134.6	51.1	124	255.2
-5.0	23	73.4	14.4	58	136.4	52.2	126	258.8
-4.4	24	75.2	15.0	59	138.2	53.3	128	262.4
-3.9	25	77.0	15.6	60	140.0	54.4	130	266.0
-3.3	26	78.8	16.7	62	143.6	55.6	132	269.6
-2.8	27	80.6	17.8	64	147.2	56.7	134	273.2
-2.2	28	82.4	18.9	66	150.8	57.8	136	276.8
-1.7	29	84.2	20.0	68	154.4	58.9	138	280.4
-1.1	30	86.0	21.1	70	158.0	60.0	140	284.0
-0.6	31	87.8	22.2	72	161.6	61.1	142	287.6
0	32	89.6	23.3	74	165.2	62.2	144	291.2
0.6	33	91.4	24.4	76	168.8	63.3	146	294.8
1.1	34	93.2	25.6	78	172.4	64.4	148	298.4

TABLE 2—*Concluded*

°C.	Temp. in °C. or °F. to be converted	°F.	°C.	Temp. in °C. or °F. to be converted	°F.	°C.	Temp. in °C. or °F. to be converted	°F.
65.6	150	302.0	93.3	200	392.0	121.1	250	482.0
66.7	152	305.6	94.4	202	395.6	123.9	255	491.0
67.8	154	309.2	95.6	204	399.2	126.7	260	500.0
68.9	156	312.8	96.7	206	402.8	129.4	265	509.0
70.0	158	316.4	97.8	208	406.4	132.2	270	518.0
71.1	160	320.0	98.9	210	410.0	135.0	275	527.0
72.2	162	323.6	100.0	212	413.6	137.8	280	536.0
73.3	164	327.2	101.1	214	417.2	140.6	285	545.0
74.4	166	330.8	102.2	216	420.8	143.3	290	554.0
75.6	168	334.4	103.3	218	424.4	146.1	295	563.0
76.7	170	338.0	104.4	220	428.0	148.9	300	572.0
77.8	172	341.6	105.6	222	431.6	154.4	310	590.0
78.9	174	345.2	106.7	224	435.2	160.0	320	608.0
80.0	176	348.8	107.8	226	438.8	165.6	330	626.0
81.1	178	352.2	108.9	228	442.4	171.1	340	644.0
82.8	180	356.0	110.0	230	446.0	176.7	350	662.0
83.3	182	359.6	111.1	232	449.6	182.2	360	680.0
84.4	184	363.2	112.2	234	453.2	187.8	370	698.0
85.6	186	366.8	113.3	236	456.8	193.3	380	716.0
86.7	188	370.4	114.4	238	460.4	198.9	390	734.0
87.8	190	374.0	115.6	240	464.0	204.4	400	752.0
88.9	192	377.6	116.7	242	467.6	210.0	410	770.0
90.0	194	381.2	117.8	244	471.2	215.6	420	788.0
91.1	196	384.8	118.9	246	474.8	221.1	430	806.0
92.2	198	388.4	120.0	248	478.4	226.7	440	824.0

TABLE 3
Properties of Water and Saturated Steam

Temp., °F.	Absolute pressure, lb./in. ²	Latent heat of evapora- tion, B.t.u./lb.	Total heat of vapor, B.t.u./lb.	Entropy, B.t.u./ (°F. × lb.)		Specific volume of vapor, ft. ³ /lb.	Density of liquid, lb./ft. ³	Viscosity of liquid, centipoises
				Liquid	Vapor			
32	0.0885	1075.8	1075.8	0.0000	2.1877	3,306	62.42	1.794
35	0.0100	1074.1	1077.1	0.0061	2.1770	2,947	62.43	1.692
40	0.1217	1071.3	1079.3	0.0162	2.1597	2,444	62.43	1.546
45	0.1475	1068.4	1081.5	0.0262	2.1429	2,036.4	62.42	1.420
50	0.1781	1065.6	1083.7	0.0361	2.1264	1,703.2	62.42	1.310
55	0.2141	1062.7	1085.8	0.0459	2.1104	1,430.7	62.40	1.213
60	0.2563	1059.9	1088.0	0.0555	2.0948	1,206.7	62.37	1.129
65	0.3056	1057.1	1090.2	0.0651	2.0796	1,021.4	62.34	1.052
70	0.3631	1054.3	1092.3	0.0745	2.0647	867.9	62.30	0.982
75	0.4298	1051.5	1094.5	0.0839	2.0502	740.0	62.26	0.919
80	0.5069	1048.6	1096.6	0.0932	2.0360	633.1	62.22	0.862
85	0.5959	1045.8	1098.8	0.1024	2.0222	543.5	62.17	0.810
90	0.6982	1042.9	1100.9	0.1115	2.0087	468.0	62.11	0.764
95	0.8153	1040.1	1103.1	0.1205	1.9955	404.3	62.06	0.721
100	0.9492	1037.2	1105.2	0.1295	1.9826	350.4	62.00	0.682
105	1.1016	1034.3	1107.3	0.1383	1.9700	304.5	61.93	0.647
110	1.2748	1031.6	1109.5	0.1471	1.9577	265.4	61.86	0.616
115	1.4709	1028.7	1111.6	0.1559	1.9457	231.9	61.79	0.586
120	1.6924	1025.8	1113.7	0.1645	1.9339	203.27	61.71	0.559
125	1.9420	1022.9	1115.8	0.1731	1.9224	178.61	61.63	0.535
130	2.2225	1020.0	1117.9	0.1816	1.9112	157.34	61.55	0.511
135	2.5370	1017.0	1119.9	0.1900	1.9002	138.95	61.46	0.490
140	2.8886	1014.1	1122.0	0.1984	1.8894	123.01	61.38	0.470
145	3.281	1011.2	1124.1	0.2066	1.8788	109.15	61.29	0.451
150	3.718	1008.2	1126.1	0.2149	1.8685	97.07	61.20	0.433
155	4.203	1005.2	1128.1	0.2230	1.8584	86.52	61.10	0.417
160	4.741	1002.3	1130.2	0.2311	1.8485	77.29	61.00	0.401
165	5.335	999.3	1132.2	0.2392	1.8388	69.19	60.90	0.386
170	5.992	996.3	1134.2	0.2472	1.8293	62.06	60.80	0.372
175	6.715	993.3	1136.2	0.2551	1.8200	55.78	60.69	0.359
180	7.510	990.2	1138.1	0.2630	1.8109	50.23	60.58	0.347
185	8.383	987.2	1140.1	0.2708	1.8020	45.31	60.47	0.336
190	9.339	984.1	1142.0	0.2785	1.7932	40.96	60.36	0.325
195	10.385	981.0	1144.0	0.2862	1.7846	37.09	60.24	0.315
200	11.526	977.9	1145.9	0.2938	1.7762	33.64	60.12	0.305

TABLE 3—Concluded

Temp., °F.	Absolute pressure, lb./in. ²	Latent heat of evapora- tion, B.t.u./lb.	Total heat of vapor, B.t.u./lb.	Entropy, B.t.u./ (°F. × lb.)		Specific volume of vapor, ft. ³ /lb.	Density of liquid, lb./ft. ³	Viscosity of liquid, centipoises
				Liquid	Vapor			
205	12.775	974.8	1147.8	0.3014	1.7680	30.57	60.00	0.295
210	14.123	971.6	1149.7	0.3090	1.7598	27.82	59.88	0.287
212	14.696	970.3	1150.4	0.3120	1.7566	26.80	59.83	0.284
215	15.599	968.4	1151.5	0.3164	1.7518	25.36	59.76	0.280
220	17.186	965.2	1153.4	0.3239	1.7440	23.15	59.63	0.272
225	18.914	962.0	1154.2	0.3313	1.7363	21.17	59.49	0.263
230	20.780	958.8	1157.0	0.3387	1.7288	19.382	59.38	0.255
235	22.795	955.5	1158.8	0.3459	1.7213	17.775	59.23	0.248
240	24.969	952.2	1160.5	0.3531	1.7140	16.323	59.10	0.242
245	27.311	948.8	1162.3	0.3604	1.7068	15.010	58.95	0.235
250	29.825	945.5	1164.0	0.3675	1.6998	13.821	58.83	0.229
260	35.429	938.7	1167.3	0.3817	1.6860	11.763	58.52	0.218
270	41.858	931.8	1170.6	0.3958	1.6727	10.061	58.24	0.208
280	49.203	924.7	1173.8	0.4096	1.6597	8.645	57.94	0.199
290	57.556	917.5	1176.8	0.4234	1.6472	7.461	57.64	0.191
300	67.013	910.1	1179.7	0.4369	1.6350	6.466	57.31	0.185
310	77.68	902.6	1182.5	0.4504	1.6231	5.626	56.98	0.179
320	89.66	894.9	1185.2	0.4637	1.6115	4.914	56.65	0.174
330	103.06	887.0	1187.7	0.4769	1.6002	4.307	56.33	
340	118.01	879.0	1190.1	0.4900	1.5891	3.788	55.96	
350	134.63	870.7	1192.3	0.5029	1.5783	3.342	55.58	
360	153.04	862.2	1194.4	0.5158	1.5677	2.957	55.22	
370	173.37	853.5	1196.3	0.5286	1.5573	2.625	54.86	
380	195.77	844.6	1198.1	0.5413	1.5471	2.335	54.47	
390	220.37	835.4	1199.6	0.5539	1.5371	2.0836	54.06	
400	247.31	826.0	1201.0	0.5664	1.5272	1.8633	53.65	

TABLE 4
Thermal Conductivity of Solids
 $k = \text{B.t.u.-ft.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.})$

Metals and Alloys			Non-Metallic Solids			
Substance	$^{\circ}\text{F.}$	k	Substance	Density lb./ft. ³	$^{\circ}\text{F.}$	k
Aluminum	32	117	Asbestos	36	32	0.087
	212	119		36	200	0.110
Brass:			Asbestos-cement board....	120	68	0.43
89-11	65	67	Boiler scale.....	—	250	1.7
82-18	65	76	Bricks:			
68-32	65	63	Alumina (fused)	—	800	1.8
Constantan—60 Cu, 40 Ni	65	13	Common	—	68	0.4
	212	16	Diatomaceous earth (molded and fused)...	38	400	0.14
Copper	32	224		38	1500	0.18
	212	218	Fireclay	—	392	0.58
Iron:				—	1832	0.95
Cast	32	32	Kaolin (insulating fire- brick)	19	392	0.050
	212	30		19	1400	0.113
Wrought	212	32	Magnesite	158	400	2.2
Lead	32	20		158	2200	1.1
	212	19	Silicon carbide	129	1112	10.7
Magnesium	32-212	92		129	2552	6.3
Monel—67 Ni, 30 Cu, 1.4 Fe, 1 Mn, 0.15 C, 0.1 Si	68	15	Concrete (stone)	—	68	0.54
Nickel	32	36	Cork, pulverized	10	32	0.021
	212	34		10	200	0.032
Silver	32	242	Corkboard	10.6	68	0.025
	212	238	Diatomaceous earth pow- der	18	100	0.039
Steel (mild)	212	26		18	1000	0.068
Tin	32	36	Glass:			
	212	34	Borosilicate	139	125	0.63
Woods metal—50 Bi, 25 Pb, 12.5 Sn, 12.5 Cd	45	8	Window	—	—	0.3-0.61
Zinc	32	65	Ice	57.5	32	1.3
	212	64	Insulating board:			
			Wood pulp	16.5	68	0.028
			Sugar cane fiber	14	68	0.028
			Magnesia, 85%	—	100	0.039
				—	400	0.046
			Mineral wool	10	68	0.023
			Rubber (soft)	—	70	0.08
			Sand, dry	94.6	68	0.19
			Sawdust	12	70	0.03
			Wood, pine:			
			Parallel to grain	34.4	70	0.020
			Cross grain	34	60	0.087
			Wool felt	20.6	86	0.03

TABLE 5
Thermal Conductivity of Liquids, Gases, and Vapors
 $k = \text{B.t.u.-ft.}/(\text{hr.} \times \text{ft.}^2 \times ^\circ\text{F.})$

Liquids			Gases and Vapors		
Substance	$^\circ\text{F.}$	k	Substance	$^\circ\text{F.}$	k
Acetic acid, 100%	68	0.099	Acetone	32	0.0057
Acetone	86	0.102		212	0.0099
Ammonia	46	0.29	Air	32	0.0140
Benzene	86	0.092		212	0.0183
Calcium chloride brine, 30% . . .	86	0.319		392	0.0226
15%	86	0.335	Ammonia	32	0.0128
Chloroform	86	0.080		212	0.0185
Ethyl acetate	68	0.101	Benzene	32	0.0052
Ethyl alcohol, 100%	68	0.105		212	0.0103
60%	68	0.176	Carbon dioxide	32	0.0085
Ethyl ether	86	0.080		212	0.0133
Gasoline	86	0.078	Chlorine	32	0.0043
Glycerol, 100%	68	0.164	Chloroform	32	0.0038
60%	68	0.220		212	0.0058
Kerosene	68	0.086	Ethane	32	0.0106
Mercury	82	4.83		212	0.0175
Sodium chloride brine, 25%	86	0.325	Ethyl alcohol	68	0.0089
12.5%	86	0.340		212	0.0124
Sulfuric acid, 90%	86	0.21	Hydrogen	32	0.100
60%	86	0.25		212	0.129
Water	32	0.320	Mercury	392	0.0197
	100	0.362	Methane	32	0.0175
	150	0.381		122	0.0215
	200	0.392	Nitrogen	32	0.0140
	250	0.396		212	0.0180
	300	0.396	Oxygen	32	0.0142
				212	0.0185
			Propane	32	0.0087
				212	0.0151
			Sulfur dioxide	32	0.0050
				212	0.0069
			Water vapor	115	0.0120
				212	0.0137
				392	0.0187

TABLE 6
Viscosity of Liquids, Gases, and Vapors

Liquids			Gases and Vapors		
Substance	Temp., °F.	Viscosity in centipoises	Substance	Temp., °F.	Viscosity in centipoises
Acetic acid.....	68	1.222	Acetone.....	32	0.00725
	122	0.796		212	0.009427
	230	0.424		415	0.01257
Acetone.....	32	0.401	Air.....	32	0.01709
	68	0.331		122	0.01951
	122	0.256		212	0.02175
Ammonia.....	-29	0.266		482	0.02770
Benzene.....	50	0.757		572	0.02946
	86	0.561		662	0.03113
	158	0.350		752	0.03277
Carbon tetrachloride....	70	0.958	Ammonia.....	32	0.00926
	96	0.793		68	0.01080
	211	0.406		212	0.01303
Chloroform.....	32	0.699	Benzene.....	32	0.00709
	104	0.464		212	0.01176
	140	0.389		415	0.01247
Decane.....	72	0.775	Carbon dioxide.....	32	0.01380
Ethyl acetate.....	32	0.583		432	0.02385
	104	0.367		576	0.02682
	158	0.279	Carbon monoxide.....	32	0.0167
Ethyl alcohol.....	32	1.772		68	0.0184
	104	0.831		212	0.0210
	158	0.510	Chlorine.....	54	0.01297
Ethyl ether.....	32	0.295		210	0.01688
	68	0.245		295	0.01897
	86	0.223	Chloroform.....	32	0.00944
Ethylene chloride.....	32	1.1322		212	0.01307
	104	0.6523		415	0.0166
	176	0.4357	Ethyl acetate.....	32	0.00684
Ethylene glycol.....	77	17.33		212	0.00943
Glycerol.....	37	4,220		415	0.0126
	68	1,069	Ethyl alcohol.....	32	0.00827
	70	777.6		212	0.01088
Hexane.....	32	0.401		415	0.01417
	104	0.271	Ethyl ether.....	32	0.00684
	140	0.229		212	0.00955
Mercury.....	-5	1.868		415	0.01234
	32	1.684	Hydrogen.....	32	0.00849
	145	1.360		215	0.01046
	309	1.092		577	0.01392
	600	0.918			

TABLE 6—*Concluded*

Liquids			Gases and Vapors		
Substance	Temp., °F.	Viscosity in centipoises	Substance	Temp., °F.	Viscosity in centipoises
Methyl alcohol.....	32	0.808	Mercury.....	523	0.0494
	104	0.449		595	0.0551
	140	0.349		716	0.0654
Nitrobenzene.....	50	2.82	Methane.....	32	0.01035
	68	1.98		68	0.01201
	95	1.55		212	0.01363
Phenol	65	12.744	Nitrogen (atmospheric).....	32	0.01695
	95	6.024		214	0.02134
	122	3.421		361	0.02464
Sulfuric acid (100%)	32	61.8	Oxygen.....	32	0.0192
	122	10.6		211	0.02485
	194	4.25		366	0.02885
Toluene.....	32	0.772	Sulfur dioxide.....	32	0.0117
	104	0.471		64	0.01242
	230	0.259		212	0.01616
Turpentine.....	32	2.248	Water vapor.....	32	0.00904
	68	1.487		86	0.01006
	104	1.071		212	0.0127
	176	0.671		405	0.0168

TABLE 7
Standard Gages

Gage number	U. S. Standard*			Birmingham Wire Gage or Stubs Gage		Brown & Sharpe Gage or American Wire Gage, thickness, in.	Std. Steel Wire Gage or Washburn & Moen Gage, thickness, in.
	Wt. of steel, lb./ft. ²	Approx. thickness,† in.	Approx. fraction, in.	Thickness, in.	Wt. of steel, lb./ft. ²		
7	7.500	0.1793	3/16	0.180	7.344	0.14428	0.1770
8	6.875	0.1644	11/64	0.165	6.732	0.12849	0.1620
9	6.250	0.1494	—	0.148	6.038	0.11443	0.1483
10	5.625	0.1345	9/64	0.134	5.467	0.10189	0.1350
11	5.000	0.1196	1/8	0.125	4.896	0.09074	0.1205
12	4.375	0.1046	7/64	0.109	4.497	0.08080	0.1055
13	3.750	0.0897	3/32	0.095	3.876	0.07196	0.0915
14	3.125	0.0747	5/64	0.083	3.386	0.06408	0.0800
15	2.812	0.0673	—	0.072	2.937	0.05706	0.0720
16	2.500	0.0598	1/16	0.065	2.651	0.05082	0.0625
17	2.250	0.0538	—	0.058	2.366	0.04525	0.0540
18	2.000	0.0478	1/20	0.049	1.999	0.04030	0.0475
19	1.750	0.0418	—	0.042	1.712	0.03589	0.0410
20	1.500	0.0359	3/80	0.035	1.428	0.03196	0.0348
21	1.375	0.0329	—	0.032	1.305	0.02846	0.0317
22	1.250	0.0299	1/32	0.028	1.142	0.02534	0.0286
23	1.125	0.0269	—	0.025	1.020	0.02257	0.0258
24	1.000	0.0239	1/40	0.022	0.897	0.02010	0.0230
25	0.875	0.0209	—	0.020	0.816	0.01790	0.0204
26	0.750	0.0179	—	0.018	0.734	0.01594	0.0181
27	0.6875	0.0164	—	0.016	0.652	0.01419	0.0173
28	0.625	0.0149	1/64	0.014	0.571	0.01264	0.0162
29	0.5625	0.0135	—	0.013	0.530	0.01125	0.0150
30	0.500	0.0120	1/80	0.012	0.489	0.01002	0.0140

* U. S. Standard Gage—Iron and steel sheets.

Birmingham or Stubs Gage—Hot & cold rolled strip, round edge flat wire, spring steel, brass, copper, seamless steel & stainless tubing and boiler tubes.

Brown & Sharpe Gage—Copper, brass, and phosphor bronze in sheets, bars, and wire.

Washburn & Moen Gage—Round steel wire in black annealed, bright basic, galvanized, tinned, and copper coated.

† $\frac{1}{8}$ " thick & heavier are classed as plates.

TABLE 8
Decimal Equivalents

Fraction	Decimal	Fraction	Decimal
1/64	0.01562	33/64	0.51562
1/32	0.03125	7/32	0.53125
3/64	0.04687	35/64	0.54687
1/16	0.06250	9/16	0.56250
5/64	0.07812	37/64	0.57812
3/32	0.09375	19/32	0.59375
7/64	0.10937	39/64	0.60937
1/8	0.12500	5/8	0.62500
9/64	0.14062	41/64	0.64062
5/32	0.15625	21/32	0.65625
11/64	0.17187	43/64	0.67187
3/16	0.18750	11/16	0.68750
13/64	0.20312	45/64	0.70312
7/32	0.21875	23/32	0.71875
15/64	0.23437	47/64	0.73437
1/4	0.25000	3/4	0.75000
17/64	0.26562	49/64	0.76562
9/32	0.28125	25/32	0.78125
19/64	0.29687	51/64	0.79687
5/16	0.31250	13/16	0.81250
21/64	0.32812	53/64	0.82812
11/32	0.34375	27/32	0.84375
23/64	0.35937	55/64	0.85937
3/8	0.37500	7/8	0.87500
25/64	0.39062	57/64	0.89062
13/32	0.40625	29/32	0.90625
27/64	0.42187	59/64	0.92187
7/16	0.43750	15/16	0.93750
29/64	0.45312	61/64	0.95312
15/32	0.46875	31/32	0.96875
31/64	0.48437	63/64	0.98437
1/2	0.50000	1	1.00000

TABLE 9
Dimensions of Standard Black and Galvanized Wrought Steel Pipe

Nominal pipe size, in.	Diameters		Wall thickness, in.	Cross-sectional area		Surface area, sq. ft. per ft. of length		Feet of pipe per sq. ft. of area	
	inside, in.	outside, in.		inside, sq. in.	outside, sq. in.	inside	outside	inside	outside
$\frac{1}{8}$	0.269	0.405	0.068	0.057	0.129	0.0705	0.106	14.199	9.431
$\frac{1}{4}$	0.364	0.540	0.088	0.104	0.229	0.0954	0.141	10.493	7.073
$\frac{3}{8}$	0.493	0.675	0.091	0.191	0.358	0.1293	0.177	7.747	5.658
$\frac{1}{2}$	0.622	0.840	0.109	0.304	0.554	0.1630	0.220	6.141	4.547
$\frac{3}{4}$	0.824	1.050	0.113	0.533	0.866	0.2158	0.275	4.635	3.637
1	1.049	1.315	0.133	0.864	1.358	0.2747	0.344	3.641	2.904
$1\frac{1}{4}$	1.380	1.660	0.140	1.495	2.164	0.3614	0.435	2.767	2.301
$1\frac{1}{2}$	1.610	1.900	0.145	2.036	2.835	0.4216	0.498	2.372	2.010
2	2.067	2.375	0.154	3.355	4.430	0.5414	0.622	1.847	1.608
$2\frac{1}{2}$	2.469	2.875	0.203	4.788	6.492	0.6476	0.753	1.547	1.328
3	3.068	3.500	0.216	7.393	9.621	0.8032	0.917	1.245	1.091
$3\frac{1}{2}$	3.548	4.000	0.226	9.886	12.566	0.9293	1.047	1.076	0.954
4	4.026	4.500	0.237	12.730	15.904	1.055	1.178	0.948	0.848
$4\frac{1}{2}$	4.506	5.000	0.247	15.947	19.635	1.181	1.308	0.847	0.763
5	5.047	5.563	0.258	20.006	24.306	1.323	1.456	0.756	0.686
6	6.065	6.625	0.280	28.891	34.472	1.590	1.734	0.629	0.576
7	7.023	7.625	0.301	38.738	45.664	1.848	2.000	0.543	0.500
8	8.071	8.625	0.277	51.161	58.426	2.114	2.258	0.473	0.442
8	7.981	8.625	0.322	50.027	58.426	2.092	2.258	0.478	0.442
9	8.941	9.625	0.342	62.786	72.762	2.342	2.520	0.427	0.396
10	10.192	10.750	0.279	81.585	90.763	2.674	2.814	0.374	0.355
10	10.136	10.750	0.307	80.691	90.763	2.660	2.814	0.376	0.355
10	10.020	10.750	0.365	78.855	90.763	2.625	2.814	0.381	0.355
11	11.000	11.750	0.375	95.033	108.434	2.882	3.076	0.347	0.325
12	12.090	12.750	0.330	114.800	127.676	3.175	3.338	0.315	0.299
12	12.000	12.750	0.375	113.097	127.676	3.145	3.338	0.318	0.299

TABLE 10
*Dimensions of Condenser and Heat Exchanger Tubing**

Outside diameter, in.	B.W.G. and Stubbs gage, No.	Wall thickness, in.	Inside diameter, in.	Cross-sectional area		Surface area, sq. ft. per ft. of length		Feet of tubing per sq. ft. of area	
				inside, sq. in.	outside, sq. in.	inside	outside	inside	outside
$\frac{1}{4}$	16	0.065	0.120	0.0114	0.0491	0.0314	0.0654	31.85	15.29
	20	0.035	0.180	0.0255	0.0491	0.0471	0.0654	21.23	15.29
	24	0.022	0.206	0.0333	0.0491	0.0539	0.0654	18.55	15.29
$\frac{3}{8}$	16	0.065	0.245	0.0471	0.1105	0.0641	0.0981	15.60	10.19
	20	0.035	0.305	0.0731	0.1105	0.0798	0.0981	12.53	10.19
	24	0.022	0.331	0.0860	0.1105	0.0866	0.0981	11.55	10.19
$\frac{1}{2}$	12	0.109	0.282	0.0624	0.1964	0.0748	0.1309	13.37	7.64
	16	0.065	0.370	0.108	0.1964	0.0969	0.1309	10.32	7.64
	20	0.035	0.430	0.145	0.1964	0.1125	0.1309	8.89	7.64
$\frac{5}{8}$	12	0.109	0.407	0.130	0.3069	0.1066	0.1636	9.38	6.11
	16	0.065	0.495	0.193	0.3069	0.1296	0.1636	7.72	6.11
	18	0.049	0.527	0.218	0.3069	0.1380	0.1636	7.25	6.11
$\frac{3}{4}$	12	0.109	0.532	0.222	0.442	0.1393	0.1963	7.18	5.09
	16	0.065	0.620	0.302	0.442	0.1623	0.1963	6.16	5.09
	18	0.049	0.652	0.334	0.442	0.1707	0.1963	5.86	5.09
$\frac{7}{8}$	12	0.109	0.657	0.338	0.602	0.1720	0.2291	5.82	4.36
	16	0.065	0.745	0.436	0.602	0.1950	0.2291	5.13	4.36
	18	0.049	0.777	0.474	0.602	0.2034	0.2291	4.91	4.36
1	12	0.109	0.782	0.481	0.786	0.2048	0.2618	4.88	3.82
	16	0.065	0.870	0.595	0.786	0.2277	0.2618	4.39	3.82
	18	0.049	0.902	0.639	0.786	0.2361	0.2618	4.23	3.82
$1\frac{1}{8}$	12	0.109	0.907	0.647	0.994	0.2375	0.2945	4.21	3.39
	16	0.065	0.995	0.778	0.994	0.2605	0.2945	3.84	3.39
	18	0.049	1.029	0.828	0.994	0.2694	0.2945	3.71	3.39
$1\frac{1}{4}$	12	0.109	1.032	0.837	1.228	0.2701	0.3271	3.70	3.05
	16	0.065	1.120	0.985	1.228	0.2932	0.3271	3.41	3.05
	18	0.049	1.152	1.043	1.228	0.3015	0.3271	3.31	3.05
$1\frac{3}{8}$	12	0.109	1.157	1.051	1.485	0.3029	0.3620	3.30	2.76
	16	0.065	1.245	1.220	1.485	0.3259	0.3620	3.07	2.76
	18	0.049	1.277	1.280	1.485	0.3343	0.3620	2.99	2.76
$1\frac{1}{2}$	12	0.109	1.282	1.290	1.767	0.3356	0.3925	2.98	2.55
	16	0.065	1.370	1.471	1.767	0.3587	0.3925	2.79	2.55
	18	0.049	1.402	1.541	1.767	0.3670	0.3925	2.73	2.55

TABLE 10—*Concluded*

Outside diameter, in.	B.W.G. and Stubbs gage, No.	Wall thickness, in.	Inside diameter, in.	Cross-sectional area		Surface area, sq. ft. per ft. of length		Feet of tubing per sq. ft. of area	
				inside, sq. in.	outside, sq. in.	inside	outside	inside	outside
1 $\frac{5}{8}$	12	0.109	1.407	1.555	2.074	0.3684	0.4254	2.71	2.35
	16	0.065	1.495	1.757	2.074	0.3914	0.4254	2.55	2.35
	18	0.049	1.527	1.829	2.074	0.3998	0.4254	2.50	2.35
1 $\frac{3}{4}$	12	0.109	1.532	1.843	2.405	0.4011	0.4582	2.49	2.18
	16	0.065	1.620	2.059	2.405	0.4241	0.4582	2.36	2.18
	18	0.049	1.652	2.146	2.405	0.4325	0.4582	2.31	2.18
1 $\frac{7}{8}$	12	0.109	1.657	2.160	2.761	0.4338	0.4909	2.31	2.04
	16	0.065	1.745	2.391	2.761	0.4568	0.4909	2.19	2.04
	18	0.049	1.777	2.477	2.761	0.4652	0.4909	2.15	2.04
2	12	0.109	1.782	2.491	3.142	0.4665	0.5233	2.14	1.91
	16	0.065	1.870	2.750	3.142	0.4896	0.5233	2.04	1.91
	18	0.049	1.902	2.837	3.142	0.4979	0.5233	2.01	1.91

* This table includes only a few of the standard gages in which tubing is made

TABLE 11
Dimensions and Weights of Copper Tubing

Nominal size, in.	Outside diameter, in.	Wall thickness of hard copper tubing				Wall thickness of soft copper tubing	
		Type K, extra heavy, in.	Type L, heavy, in.	Type M, standard, in.	Type O, light, in.	Type K, extra heavy, in.	Type L, heavy, in.
1/4	3/8	0.032	0.030	0.025	—	0.032	0.030
3/8	1/2	0.049	0.035	0.025	—	0.049	0.035
1/2	5/8	0.049	0.040	0.028	—	0.049	0.040
3/4	7/8	0.065	0.045	0.032	—	0.065	0.045
1	1 1/8	0.065	0.050	0.035	—	0.065	0.050
1 1/4	1 3/8	0.065	0.055	0.042	—	0.065	0.055
1 1/2	1 5/8	0.072	0.060	0.049	—	0.072	0.060
2	2 1/8	0.083	0.070	0.058	—	0.083	0.070
2 1/2	2 5/8	0.095	0.080	0.065	—	0.095	0.080
3	3 1/8	0.109	0.090	0.072	0.049	0.109	0.090
3 1/2	3 5/8	0.120	0.100	0.083	0.049	0.120	0.100
4	4 1/8	0.134	0.110	0.095	0.068	0.134	0.110
5	5 1/8	0.160	0.125	0.109	0.065	0.160	0.125
6	6 1/8	0.192	0.140	0.122	0.072	0.192	0.140
8	8 1/8	0.271	0.200	0.170	0.083	0.271	0.200
10	10 1/8	0.338	0.250	0.212	0.109	0.338	0.250
12	12 1/8	0.405	0.280	0.254	0.134	0.405	0.280

TABLE 12
Tap Drill Sizes for American Standard Taper Pipe Threads

Nominal pipe size, in.	Threads per in.	Tap drill size	Dec. equiv. of drill, in.
$\frac{1}{8}$	27	R	0.339
$\frac{1}{4}$	18	$7/16''$	0.437
$\frac{3}{8}$	18	$37/64''$	0.578
$\frac{1}{2}$	14	$23/32''$	0.719
$\frac{3}{4}$	14	$59/64''$	0.921
1	$11\frac{1}{2}$	$1-5/32''$	1.156
$1\frac{1}{4}$	$11\frac{1}{2}$	$1-1/2''$	1.500
$1\frac{1}{2}$	$11\frac{1}{2}$	$1-47/64''$	1.734
2	$11\frac{1}{2}$	$2-7/32''$	2.218

TABLE 13
Tap Drill Sizes for American National Form Threads

Nominal size and threads per in.*	Tap drill size for approx. 75% full threads	Decimal equivalent of tap drill, in.	Nominal size and threads per in.	Tap drill size for approx. 75% full threads	Decimal equivalent of tap drill, in.
0-80NF	$3/64''$	0.0469	$5/32''-32NS$	$1/8''$	0.1250
1-64NC	53	0.0595	$5/32''-36NS$	30	0.1285
1-72NF	53	0.0595	$11/64''-32NS$	$9/64''$	0.1406
2-56NC	50	0.0700	$3/16''-24NS$	26	0.1470
2-64NF	50	0.0700	$3/16''-32NS$	22	0.1570
3-48NC	47	0.0785	$13/64''-24NS$	20	0.1610
3-56NF	45	0.0820	$7/32''-24NS$	16	0.1770
4-40NC	43	0.0890	$7/32''-32NS$	12	0.1890
4-48NF	42	0.0935	$15/64''-24NS$	10	0.1935
5-40NC	38	0.1015	$1/4''-20NC$	7	0.2010
5-44NF	37	0.1040	$1/4''-28NF$	3	0.2130
6-32NC	36	0.1065	$5/16''-18NC$	F	0.2570
6-40NF	33	0.1130	$5/16''-24NF$	I	0.2720
8-32NC	29	0.1360	$3/8''-16NC$	$5/16''$	0.3125
8-36NF	29	0.1360	$3/8''-24NF$	Q	0.3320
10-24NC	25	0.1495	$7/16''-14NC$	U	0.3680
10-32NF	21	0.1590	$7/16''-20NF$	$25/64''$	0.3906
12-24NC	16	0.1770	$1/2''-13NC$	$27/64''$	0.4219
12-28NF	14	0.1820	$1/2''-20NF$	$29/64''$	0.4531
$1/16''-64NS$	$3/64''$	0.0469	$9/16''-12NC$	$31/64''$	0.4844
$1/16''-72NS$	$3/64''$	0.0469	$9/16''-18NF$	$33/64''$	0.5156
$5/64''-60NS$	$1/16''$	0.0625	$5/8''-11NC$	$17/32''$	0.5312
$5/64''-72NS$	52	0.0635	$5/8''-18NF$	$37/64''$	0.5781
$3/32''-48NS$	49	0.0730	$3/4''-10NC$	$21/32''$	0.6562
$3/32''-50NS$	49	0.0730	$3/4''-16NF$	$11/16''$	0.6875
$7/64''-48NS$	43	0.0890	$7/8''-9NC$	$49/64''$	0.7656
$1/8''-32NS$	$3/32''$	0.0937	$7/8''-14NF$	$13/16''$	0.8125
$1/8''-40NS$	38	0.1015	$1''-8NC$	$7/8''$	0.8750
$9/64''-40NS$	32	0.1160	$1''-14NF$	$15/16''$	0.9375

* NC—Std. Coarse Threads; NF—Std. Fine Threads; NS—Special Threads.

TABLE 14
U. S. Standard Screen-scale Sieves

Sieve No.	Meshes per lineal in.	Sieve opening		Wire diameter	
		in.	mm.	in.	mm.
2½	2.58	0.315	8.00	0.073	1.85
3	3.03	0.265	6.73	0.065	1.65
3½	3.57	0.223	5.66	0.057	1.45
4	4.22	0.187	4.76	0.050	1.27
5	4.98	0.157	4.00	0.044	1.12
6	5.81	0.132	3.36	0.040	1.02
7	6.80	0.111	2.83	0.036	0.92
8	7.89	0.0937	2.38	0.0331	0.84
10	9.21	0.0787	2.00	0.0299	0.76
12	10.72	0.0661	1.68	0.0272	0.69
14	12.58	0.0555	1.41	0.0240	0.61
16	14.66	0.0469	1.19	0.0213	0.54
18	17.15	0.0394	1.00	0.0189	0.48
20	20.16	0.0331	0.84	0.0165	0.42
25	23.47	0.0280	0.71	0.0146	0.37
30	27.62	0.0232	0.59	0.0130	0.33
35	32.15	0.0197	0.50	0.0114	0.29
40	38.02	0.0165	0.42	0.0098	0.25
45	44.44	0.0138	0.35	0.0087	0.22
50	52.36	0.0117	0.297	0.0074	0.188
60	61.93	0.0098	0.250	0.0064	0.162
70	72.46	0.0083	0.210	0.0055	0.140
80	85.47	0.0070	0.177	0.0047	0.119
100	101.01	0.0059	0.149	0.0040	0.102
120	120.48	0.0049	0.125	0.0034	0.086
140	142.86	0.0041	0.105	0.0029	0.074
170	166.67	0.0035	0.088	0.0025	0.063
200	200.00	0.0029	0.074	0.0021	0.053
230	238.10	0.0024	0.062	0.0018	0.046
270	270.26	0.0021	0.053	0.0016	0.041
325	323.00	0.0017	0.044	0.0014	0.036

TABLE 15
Tyler Standard Screen-scale Sieves

Mesh	Sieve opening		Wire diameter	
	in.	mm.	in.	mm.
—	1.050	26.67	0.148	3.76
—	0.883	22.43	0.135	3.43
—	0.742	18.85	0.135	3.43
—	0.624	15.85	0.120	3.05
—	0.525	13.33	0.105	2.67
—	0.441	11.20	0.105	2.67
—	0.371	9.423	0.092	2.34
2½	0.312	7.925	0.088	2.24
3	0.263	6.680	0.070	1.78
3½	0.221	5.613	0.065	1.65
4	0.185	4.699	0.065	1.65
5	0.156	3.962	0.044	1.12
6	0.131	3.327	0.036	0.91
7	0.110	2.794	0.0328	0.833
8	0.093	2.362	0.032	0.81
9	0.078	1.981	0.033	0.84
10	0.065	1.651	0.035	0.89
12	0.055	1.397	0.028	0.71
14	0.046	1.168	0.025	0.64
16	0.0390	0.991	0.0235	0.597
20	0.0328	0.833	0.0172	0.437
24	0.0276	0.701	0.0141	0.358
28	0.0232	0.589	0.0125	0.318
32	0.0195	0.495	0.0118	0.300
35	0.0164	0.417	0.0122	0.310
42	0.0138	0.351	0.0100	0.254
48	0.0116	0.295	0.0092	0.234
60	0.0097	0.246	0.0070	0.178
65	0.0082	0.208	0.0072	0.183
80	0.0069	0.175	0.0056	0.142
100	0.0058	0.147	0.0042	0.107
115	0.0049	0.124	0.0038	0.097
150	0.0041	0.104	0.0026	0.066
170	0.0035	0.088	0.0024	0.061
200	0.0029	0.074	0.0021	0.053

TABLE 16
Approximate Net Price of Standard Black Wrought Steel Pipe and Black Malleable Iron Screwed Fittings.

Size, in.	Pipe, price per 100 ft.	Price per 100													
		Elbows			Street elbows		Tees		Caps	Reducers	Bushings		Couplings	Unions	
		90°	90° red.	45°	90°	45°	straight	red.			hexagon	face		gasket type	ground-joint type
1/8	\$3.53	\$3.66	—	\$4.39	\$4.39	—	\$5.12	—	\$2.20	—	—	—	\$3.06	\$10.80	\$10.24
1/4	3.63	3.66	\$4.39	4.39	4.39	—	5.12	\$5.86	2.20	\$3.66	\$2.35	\$4.70	3.06	10.80	10.24
3/8	3.63	3.92	5.12	5.12	5.12	\$5.85	5.60	6.49	2.93	3.66	2.35	4.70	3.67	12.00	11.52
1/2	4.66	4.39	5.86	6.59	5.86	7.32	5.86	8.05	3.66	4.39	2.35	5.29	4.28	13.20	12.80
3/4	5.87	5.60	8.05	8.78	8.78	10.23	7.28	10.98	5.86	6.59	2.94	6.47	6.12	16.20	15.36
1	8.26	9.52	10.25	11.71	10.98	13.18	12.44	13.91	7.32	8.78	3.53	7.64	7.96	19.80	20.48
1 1/4	11.18	14.64	16.10	17.57	16.84	20.50	19.03	21.23	9.52	10.98	4.12	10.00	10.40	27.60	30.72
1 1/2	13.37	18.30	20.50	21.96	21.23	25.60	24.16	27.08	13.18	13.18	5.29	12.44	12.85	34.80	39.68
2	17.99	29.28	32.21	35.14	33.67	40.20	38.06	41.72	18.30	20.50	8.23	18.82	17.14	45.00	58.88
2 1/2	28.44	52.70	58.56	62.95	60.76	—	65.88	73.20	29.28	36.60	12.35	28.22	24.48	93.00	97.28
3	37.19	73.20	80.52	87.84	84.18	—	95.15	106.14	43.92	54.90	17.64	41.16	36.72	126.00	140.80
3 1/2	47.84	109.90	120.50	131.80	—	—	146.40	161.00	58.55	80.50	23.55	70.60	55.00	220.00	—
4	46.68	131.76	146.40	161.04	153.70	—	175.68	193.98	80.52	102.48	29.40	88.20	61.20	260.00	—
5	76.96	238.00	—	285.40	—	—	311.00	344.00	131.80	—	54.65	153.90	—	—	—
6	99.84	340.50	—	409.80	—	—	439.50	483.00	197.60	—	73.55	220.50	—	—	—

TABLE 17
Approximate Net Price of Standard Galvanized Wrought Steel Pipe and Galvanized Malleable Iron Screwed Fittings

Size, in.	Pipe, price per 100 ft.	Price per 100													
		Elbows			Street elbows		Tees		Caps	Reducers	Bushings		Couplings	Unions	
		90°	90° red.	45°	90°	45°	straight	red.			hexagon	face		gasket type	ground-joint type
1/8	\$4.90	\$5.12	—	\$5.86	\$5.86	—	\$6.59	—	\$2.93	—	—	\$3.67	\$16.20	\$11.52	
1/4	4.79	5.12	\$5.86	5.86	5.86	\$6.59	6.59	\$7.32	2.93	\$4.13	\$6.19	3.67	16.20	11.52	
3/8	4.79	5.04	6.59	6.59	6.59	7.32	7.28	8.78	3.66	4.13	6.19	4.90	18.00	12.80	
1/2	5.67	5.96	7.32	8.78	7.32	9.22	7.32	10.25	5.12	4.13	7.22	6.12	19.80	14.08	
3/4	7.10	7.28	10.25	11.71	11.71	13.18	9.52	14.64	7.32	5.16	8.77	7.96	24.00	17.92	
1	9.96	12.44	13.18	15.37	14.64	16.84	16.10	18.30	9.52	6.19	10.32	11.02	30.00	24.32	
1 1/4	13.48	19.03	21.23	22.69	21.96	26.35	24.89	27.82	12.44	7.22	12.90	15.30	42.00	35.84	
1 1/2	16.12	24.16	26.35	28.55	27.82	33.67	31.48	35.14	16.84	9.29	17.03	19.58	54.00	46.08	
2	21.69	38.06	41.72	45.38	43.92	52.70	49.78	54.90	24.16	14.44	24.77	24.48	69.00	66.56	
2 1/2	34.29	69.54	76.86	80.52	80.52	—	87.84	95.16	38.06	21.67	37.10	33.66	141.00	115.20	
3	44.84	95.16	106.14	113.46	109.80	—	124.44	139.08	57.10	30.96	54.05	48.96	189.00	168.96	
3 1/2	57.61	143.00	157.50	172.10	—	—	190.02	208.50	76.90	41.20	92.65	70.40	330.00	—	
4	68.26	172.02	190.32	208.62	201.30	—	226.92	252.54	106.14	51.60	115.95	85.68	390.00	—	
5	92.68	311.00	—	373.20	—	—	406.00	446.30	172.00	95.30	201.00	—	—	—	
6	120.24	443.00	—	534.50	—	—	571.00	629.50	256.20	128.90	288.30	—	—	—	

TABLE 18

Approximate Net Price of Standard Black Wrought Steel Right Hand Nipples

Size, in.	Length, in.		Price per 100							
			Length, in.							
	close	short	close	short	2	2½	3	3½	4	4½
⅛	¾	1½	\$1.64	\$1.97	\$2.02	\$2.30	\$2.30	\$2.59	\$2.59	\$2.88
¼	7⁄8	1½	1.64	1.97	2.02	2.30	2.30	2.59	2.59	2.88
⅜	1	1½	1.64	1.97	2.02	2.30	2.30	2.59	2.59	2.88
½	1⅛	1½	1.97	2.30	2.30	2.59	2.59	2.88	2.88	3.17
¾	1⅜	2	2.62	2.95	—	2.88	3.17	3.46	3.74	4.03
1	1½	2	3.28	3.61	—	4.03	4.03	4.32	4.61	4.90
1¼	1⅝	2½	4.26	4.92	—	—	4.61	5.18	5.76	6.43
1½	1¾	2½	5.25	5.90	—	—	5.76	6.34	7.20	7.78
2	2	2½	6.89	7.87	—	—	8.06	8.93	8.79	10.66
2½	2½	3	12.79	14.76	—	—	—	16.70	18.14	19.58
3	2⅝	3	17.71	19.35	—	—	—	18.72	21.02	23.33
3½	2¾	4	23.62	32.14	—	—	—	—	—	30.53
4	2⅞	4	30.18	36.74	—	—	—	—	—	36.86
4½	2⅞	4	44.28	52.48	—	—	—	—	—	50.04
5	3	4½	52.48	65.60	—	—	—	—	—	—
6	3⅛	4½	65.60	80.36	—	—	—	—	—	—

Size, in.	Length, in.								
	5	5½	6	7	8	9	10	11	12
⅛	\$3.17	\$3.46	\$3.74	\$4.03	\$4.32	\$4.61	\$4.90	\$5.18	\$5.76
¼	3.17	3.46	3.74	4.03	4.32	4.61	4.90	5.18	5.76
⅜	3.17	3.46	3.74	4.03	4.32	4.61	4.90	5.18	5.76
½	3.46	3.74	4.03	4.32	4.90	5.47	6.05	6.62	7.20
¾	4.32	4.61	4.90	5.47	6.05	6.62	7.20	7.78	8.35
1	5.18	5.47	5.76	6.91	7.78	8.64	9.22	10.08	10.94
1¼	6.91	7.49	8.06	9.22	10.37	11.52	12.67	13.82	14.98
1½	8.35	8.93	9.79	10.94	12.38	13.54	14.98	16.13	17.28
2	11.52	12.67	13.82	15.26	16.99	19.01	22.46	23.04	25.06
2½	21.02	22.46	24.48	27.07	29.66	32.26	35.14	37.73	40.32
3	25.63	27.65	29.95	34.27	38.59	43.20	47.52	51.84	56.16
3½	32.83	34.85	37.15	41.76	46.37	50.69	55.30	59.90	64.51
4	39.74	42.62	46.08	51.84	57.60	63.36	69.12	74.88	80.64
4½	54.72	60.48	64.80	72.00	79.20	86.40	93.60	100.80	108.00
5	59.04	62.50	65.95	75.17	83.81	90.72	97.63	104.83	111.74
6	74.30	78.62	83.52	96.77	105.98	117.50	124.13	132.84	142.27

TABLE 19

Approximate Net Price of Standard Galvanized Wrought Steel Right Hand Nipples

Size, in.	Length, in.		Price per 100							
			Length, in.							
	close	short	close	short	2	2½	3	3½	4	4½
⅛	¾	1½	\$1.79	\$2.05	\$2.16	\$2.38	\$2.59	\$2.81	\$3.02	\$3.24
¼	7⁄8	1½	1.79	2.05	2.16	2.38	2.59	2.81	3.02	3.24
⅜	1	1½	1.79	2.05	2.16	2.38	2.59	2.81	3.02	3.24
½	1⅛	1½	2.05	2.30	2.38	2.59	2.81	3.02	3.24	3.46
¾	1⅜	2	2.56	3.07	—	2.81	3.02	3.46	3.89	4.32
1	1½	2	3.33	3.94	—	3.89	4.32	4.75	5.18	5.62
1¼	1⅝	2½	4.35	5.63	—	—	5.62	6.04	6.48	7.34
1½	1¾	2½	5.63	6.66	—	—	6.48	7.34	8.21	9.07
2	2	2½	7.68	8.96	—	—	8.64	9.72	10.80	11.88
2½	2½	3	15.36	16.90	—	—	—	16.20	17.98	19.44
3	2⅝	3	20.48	22.27	—	—	—	21.82	23.76	25.92
3½	2¾	4	29.44	35.84	—	—	—	—	—	33.05
4	2⅞	4	32.51	40.96	—	—	—	—	—	39.96
4½	2⅞	4	44.80	55.04	—	—	—	—	—	50.76
5	3	4½	58.62	73.47	—	—	—	—	—	—
6	3⅛	4½	73.22	91.14	—	—	—	—	—	—

Size, in.	Length, in.								
	5	5½	6	7	8	9	10	11	12
⅛	\$3.46	\$3.67	\$4.10	\$4.75	\$5.40	\$6.05	\$6.48	\$7.13	\$9.22
¼	3.46	3.67	4.10	4.75	5.40	6.05	6.48	7.13	9.22
⅜	3.46	3.67	4.10	4.75	5.40	6.05	6.48	7.13	9.22
½	3.67	3.89	4.32	4.97	5.62	6.26	6.91	7.56	9.25
¾	4.75	5.18	5.62	6.26	6.91	7.56	8.42	9.07	9.72
1	6.04	6.48	7.34	8.21	9.07	9.94	10.80	11.66	12.53
1¼	7.99	8.64	9.50	10.37	12.10	13.39	14.69	15.98	17.28
1½	9.72	10.37	11.23	12.53	14.04	15.77	17.28	18.79	20.18
2	12.96	14.04	15.12	17.06	19.01	20.95	22.90	24.81	26.78
2½	21.17	22.90	24.62	27.65	31.10	34.56	38.02	41.47	44.93
3	28.08	30.24	32.40	36.72	41.04	45.36	49.68	54.00	58.32
3½	35.64	38.45	41.26	46.66	51.84	57.46	62.86	68.26	73.66
4	43.20	46.44	49.68	56.16	62.64	69.12	75.60	82.08	88.56
4½	55.08	59.40	64.80	73.44	82.08	90.72	99.36	108.00	116.64
5	65.02	69.34	73.44	85.32	93.96	102.17	110.60	119.23	127.44
6	81.86	87.05	92.88	108.00	119.52	130.25	141.05	152.28	162.86

TABLE 20

Approximate Net Price of Brass and Iron Valves and Iron Cocks

Size, in.	Brass						Iron Body		Iron
	Globe or Angle		Gate		Check		std. gate	swing check	std. cock
	std. 125 lb.	low press. 100 lb.	std. 125 lb.	low press. 100 lb.	std. 125 lb.	low press. 100 lb.			
1/8	\$0.95	—	—	—	\$0.84	—	—	—	—
1/4	0.95	—	\$1.17	\$0.56	0.84	—	—	—	—
3/8	1.00	\$0.42	1.17	0.56	0.92	\$0.45	—	—	—
1/2	1.27	0.53	1.27	0.60	1.14	0.55	—	—	—
3/4	1.58	0.66	1.61	0.77	1.45	0.70	—	—	\$1.03
1	1.90	0.79	2.15	1.02	1.69	0.82	—	—	1.27
1 1/4	2.64	1.10	2.95	1.39	2.38	1.15	—	—	1.57
1 1/2	3.70	1.54	3.90	1.86	3.33	1.61	—	—	1.91
2	5.60	2.34	5.37	2.55	5.02	2.43	\$9.08	\$8.98	2.65

TABLE 21

Approximate Net Price of Brass Solder-Joint Fittings

Fitting	Price per 100						
	Nominal size, in.						
	3/8	1/2	3/4	1	1 1/4	1 1/2	2
<i>Elbows, 90° (straight or reducing):</i>							
Copper to copper.....	\$7.50	\$9.00	\$15.00	\$27.50	\$42.00	\$57.50	\$82.50
Copper to outside I.P.S.....	7.50	9.00	15.00	27.50	42.00	57.50	82.50
Copper to inside I.P.S.....	7.50	9.00	15.00	27.50	42.00	57.50	82.50
<i>Elbows, 45°:</i>							
Copper to copper.....	7.50	9.00	15.00	27.50	42.00	57.50	82.50
<i>Street elbows, 90°:</i>							
Copper to copper.....	7.50	9.00	15.00	27.50	42.00	57.50	82.50
<i>Street elbows, 45°:</i>							
Copper to copper.....	7.50	9.00	15.00	27.50	42.00	57.50	82.50
<i>Tee (straight or reducing):</i>							
Copper to copper.....	11.50	13.50	24.50	38.00	57.50	82.50	115.00
Copper to I.P.S.....	11.50	13.50	24.50	38.00	57.50	82.50	115.00
<i>Unions:</i>							
Copper to copper.....	38.00	45.60	65.00	82.50	115.00	167.50	210.00
Copper to outside I.P.S.....	52.50	65.00	75.00	110.00	175.00	202.50	285.00
<i>Couplings (straight or reducing):</i>							
Copper to copper.....	4.50	6.00	9.00	12.00	15.00	19.00	34.00
Copper to outside I.P.S.....	7.50	9.00	13.50	23.00	38.00	52.50	75.00
Copper to inside I.P.S.....	7.50	9.00	13.50	23.00	38.00	52.50	75.00
<i>Bushings:</i>							
Copper to inside I.P.S. reduced by one size.....	4.50	6.00	9.00	12.00	15.00	19.00	34.00
<i>Plugs.....</i>	4.50	6.00	9.00	12.00	38.00	52.50	75.00
<i>Caps.....</i>	4.50	6.00	9.00	12.00	38.00	52.50	75.00

TABLE 22
Approximate Net Price of 85 Per Cent Magnesia Pipe Insulation

Nominal pipe size, in.	Thickness of insulation, in.	Price per ft.
$\frac{1}{2}$	$\frac{7}{8}$	\$0.13
$\frac{3}{4}$	$\frac{7}{8}$	0.14
1	$\frac{7}{8}$	0.15
$1\frac{1}{4}$	$\frac{7}{8}$	0.17
$1\frac{1}{2}$	$\frac{7}{8}$	0.19
2	$1\frac{1}{2}$	0.20
$2\frac{1}{2}$	$1\frac{1}{2}$	0.22
3	$1\frac{1}{2}$	0.25
$3\frac{1}{2}$	$1\frac{1}{2}$	0.28
4	$1\frac{1}{8}$	0.33
$4\frac{1}{2}$	$1\frac{1}{8}$	0.36
5	$1\frac{1}{8}$	0.39
6	$1\frac{1}{8}$	0.44
7	$1\frac{1}{4}$	0.55
8	$1\frac{1}{4}$	0.61
9	$1\frac{1}{4}$	0.66
10	$1\frac{1}{4}$	0.72
11	$1\frac{1}{4}$	0.88
12	$1\frac{1}{2}$	1.19
14	$1\frac{1}{2}$	1.57

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